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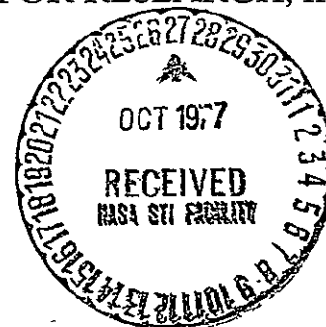
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THE UNIVERSITY OF KANSAS CENTER FOR RESEARCH, INC.

2291 Irving Hill Drive—Campus West
Lawrence, Kansas 66045



Progress Report

for

A RESEARCH PROGRAM TO REDUCE INTERIOR
NOISE IN GENERAL AVIATION AIRPLANES

KU-FRL-317-5

NASA Grant NSG 1301

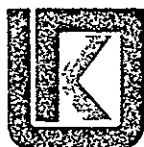
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Table of Contents

	<u>Page</u>
List of Symbols.	ii
1. Introduction.	1
2. Description of the Test Facility.	2
3. The Influence of Stiffness.	7
3.1 Analytical Considerations.	7
3.2 Experimental Results	8
3.3 Comparison	8
4. The Influence of Mass	11
4.1 Analytical Considerations.	11
4.2 Experimental Results	11
4.3 Comparison	12
5. The Influence of Pressurization	17
5.1 Analytical Considerations.	17
5.2 Experimental Results	17
5.3. Comparison	18
6. The Influence of Vibration Damping Material	24
6.1 Experimental Results	24
7. Test Results.	28
8. Conclusions	70
References	71
Appendix A. Vibration Damping Materials	A-1

List of Symbols

<u>Symbol</u>	<u>Definition</u>	<u>Dimension</u>
a	Panel dimension	m
b	Panel dimension	m
c	Speed of sound	m/s
D	Flexural rigidity	Nm
E	Young's modulus	N/m ²
f	Frequency	Hz
K	Constant	-
$\bar{m}$	Panel surface mass	kg/m ²
N	Internal membrane force per unit length	N/m ²
P	Pressure	N/m ²
s	Fraction of surface mass which participates fully in the motion at resonance	-
TL	Transmission loss	dB
t	Thickness of panel	m

Greek Symbols

Δ	Difference	-
ρ	Density	kg/m ³
ν	Poission's ratio	-
ω	Frequency	rad./s
θ	Angle of incidence of sound	-

1. Introduction

The objective of this NASA-sponsored KU-FRL noise project (NSG 1301) is to investigate experimentally and analytically the transmission of sound through isolated panels. The purpose of this report is to give the first test results of the panels tested up to mid-September.

In the month of August the KU-FRL noise research team completed the construction and the calibration of the test facility and started testing flat panels. During this time the team also continued predictions of behavior of the panels according to pertinent analytical methods. The first results have permitted improvement of semi-empirical methods of predicting panel transmission loss.

In this report the influence of stiffness and mass of the plate will be discussed. Also the effect of pressurization and the usefulness of damping materials will be described.

Gratitude is expressed to Beech, Cessna and Grumman American Aircraft Corporations for providing most of the panels, and some of the panel treatments, tested for this report. Appreciation is also expressed to Insul-Coustics Corporation and Specialty Composites Corporation for providing some of the vibration damping materials tested.

2. Description of the Test Facility

The KU-FRL test facility for measurement of sound transmission through panels is described in detail in Reference 1. This chapter will give only a brief description. A sketch of the test facility is shown in Figure 2.1, and a photograph in Figure 2.2.

The panel to be tested is mounted between the two chambers A and B. The source chamber A, consisting of a massive brick wall, concrete collar and steel box, contains nine evenly spaced loudspeakers. This chamber can be considered a speaker box. Its purpose is to support the speakers and to prevent radiation of sound to the rear and side. It contains sound absorbing materials to minimize standing waves that could induce undesired speaker sound radiation characteristics.

The panel under test is separated from the front side of the speaker baffle by a small distance, about one inch. This arrangement prevents standing waves between the baffle and specimen at frequencies in the range of interest, i.e. 20-5000 Hz. Other standing waves, parallel to the panel and the speaker baffle, would disturb the desired uniformity of excitation at the panel surface. The strength of these standing waves is reduced by sound absorbing material, which nearly fills the space between baffle and panel.

The receiving chamber (B in Figure 2.1) is an acoustic termination which absorbs very nearly all the sound passing through the panel. It also significantly improves the noise environment in which research personnel work; and at the same time, it influences the transmission of sound through the panel in the same way as an infinite space of air. To facilitate the installation of test specimens between this termination and the speaker box, the receiving chamber is mounted on wheels and rests on a steel table.

The loudspeakers can be driven by the amplified signal of either a white noise generator, a pure tone generator, or a tape recording of in-flight boundary layer pressure fluctuations. An equalizer is included in this noise generating system to obtain a reasonably flat noise spectrum.

The noise measuring system includes two microphones, one on each side of the panel under test. The output signals of these microphones are fed, one at a time, into a real time analyzer. The resulting spectra are plotted by an X-Y recorder. Next, these curves are read into a desktop calculator having curve digitizing capabilities, which subtracts one spectrum from the other, applies corrections and plots the final test results. The corrections account for sound energy reflected from the panel and for the different acoustic impedances on each side of pressurized panels.

For testing under simulated pressurized-airplane conditions, there is a depressurizing system to reduce the pressure in chamber A below the atmospheric pressure of chamber B.

Figure 2.3 shows details and dimensions of the test section and the clamping system. Each test panel is 20 in. x 20 in. However, the effective panel size is reduced to 18 in. x 18 in. by the one inch clamping margin on all edges. Each of the eight clamps is tightened with a torque wrench to 25 in.lb. torque. This corresponds to approximately 390 lb. force in each clamp, or a total clamping force of 3100 lb. on the panel edges.

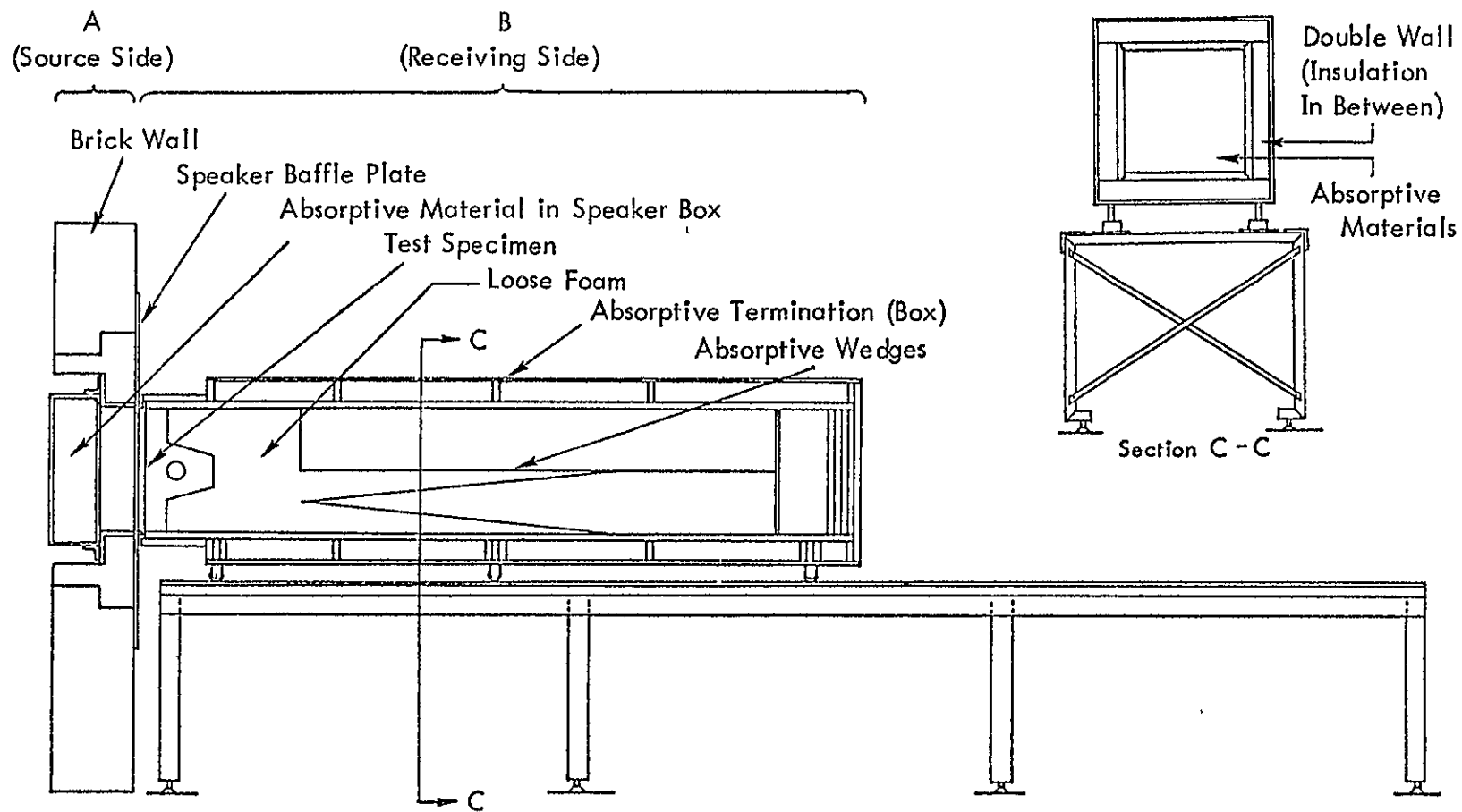


Figure 2.1: The plane wave tube

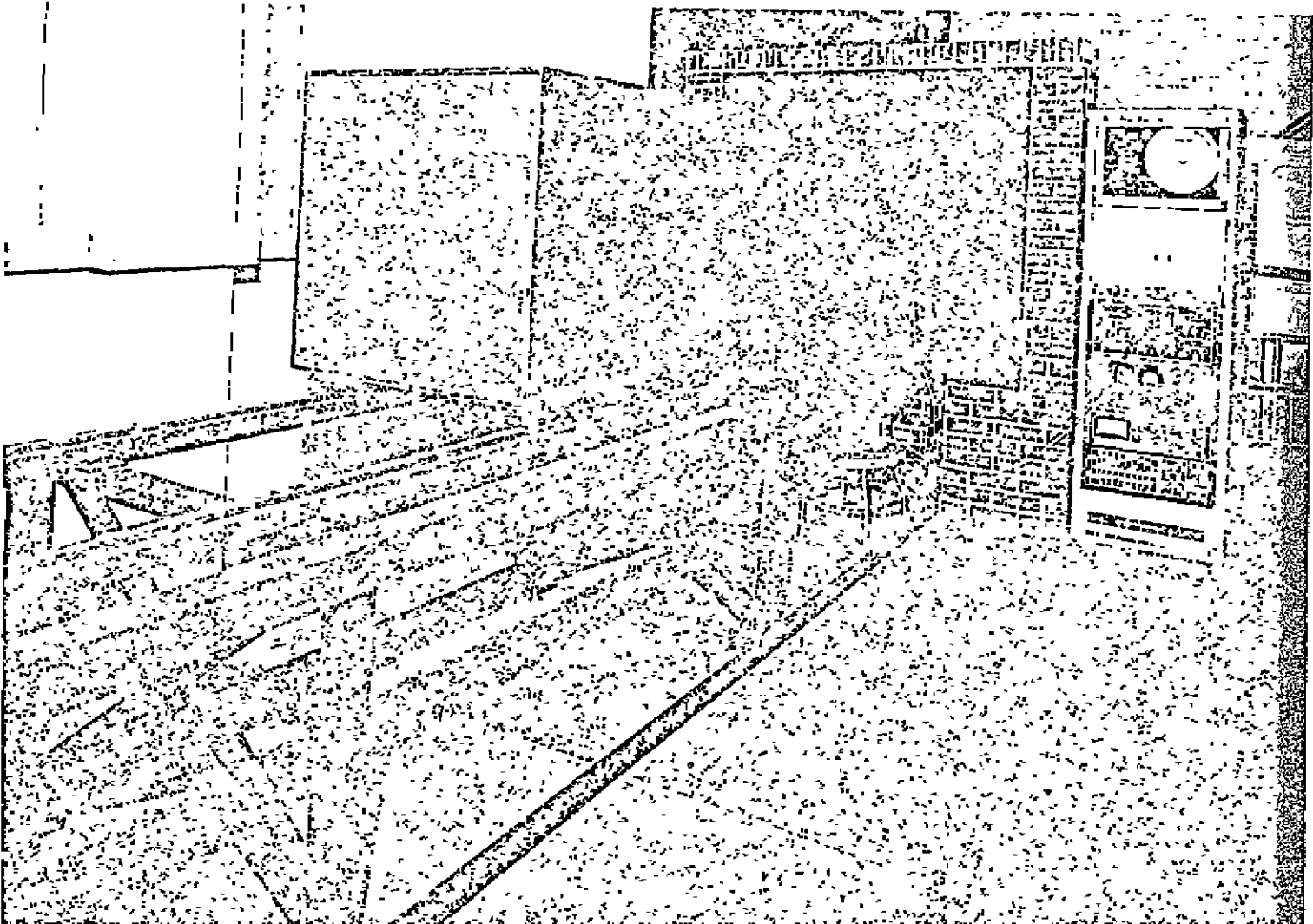
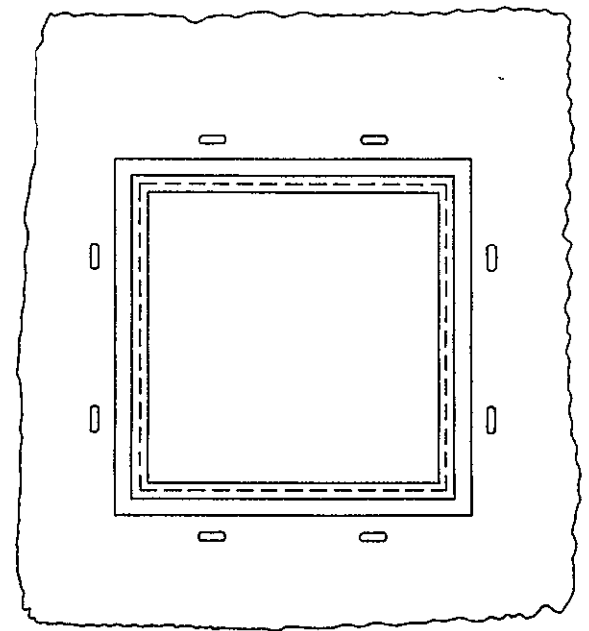
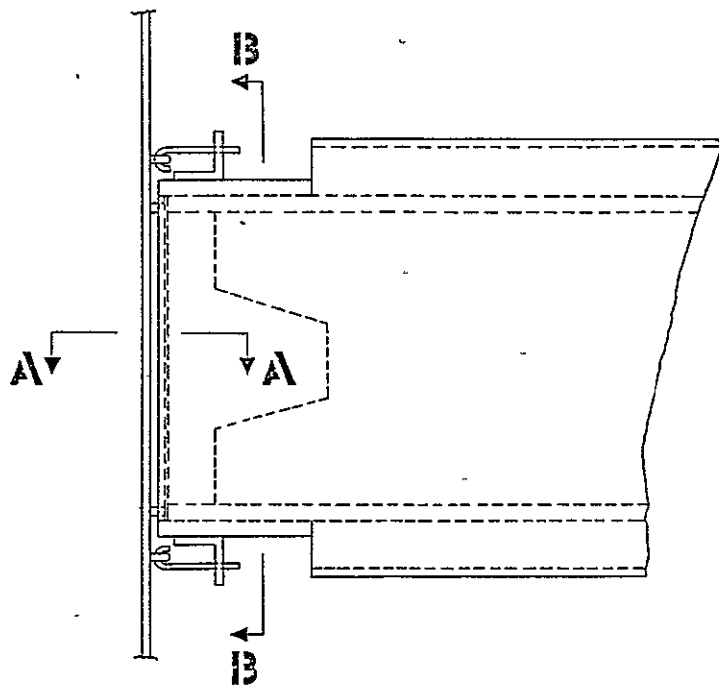
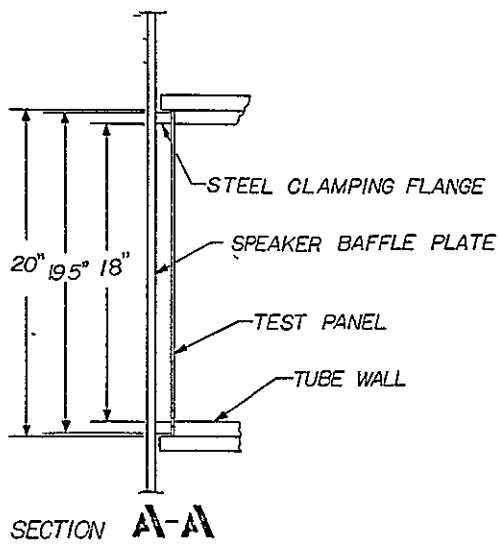


Figure 2.2: The plane wave tube test facility



SECTION B-B

Figure 3.3: Detail of the panel clamping system.

3. The Influence of Stiffness

This chapter contains a discussion of the influence of the stiffness of a panel on its vibrations and transmission loss. Analytical and semi-empirical methods are included, and experimental results are given and compared with theory.

3.1 Analytical Consideration

Below the fundamental frequency the panel motion and noise transmission are controlled by stiffness. In general aviation aircraft, fundamental panel frequencies range from about 50 to 200 Hz, depending on panel size, material and edge conditions.

Reference 2 gives the following equation to estimate the

transmission loss (TL) at frequency $\frac{f_{1,1}}{4}$:

$$TL \left(\frac{f_{1,1}}{4}, \text{stiffness} \right) = TL \left(f_{1,1}, 45^\circ, \text{mass law} \right) + 10 \log s^2 + 15 \quad [\text{dB}] \quad (3.1)$$

where $f_{1,1}$ = the fundamental resonance frequency

s = fraction of the total surface mass of the plate which participates fully in the motion at resonance ($s \approx 0.2$)

$TL \left(f_{1,1}, 45^\circ, \text{mass law} \right)$ = the mass law TL for 45° incidence at the resonance frequency, considering the total surface mass.

The KU-FRL noise research team derived the following semi-empirical equation to predict the transmission loss at a certain frequency, f , below the fundamental resonance frequency:

$$TL = 20 \log \frac{\bar{m} (f_{1,1}^2 - f^2)}{f} - 39 \quad [\text{dB}] \quad (3.2)$$

where $\bar{m}$ = the panel surface mass $\left(\frac{\text{kg}}{\text{m}^2} \right)$

For the derivation of this equation, see Reference 3.

One possible way to get a high transmission loss in the sub-resonance region is to raise the resonance frequency by rigidification of the panel. This effect can also be achieved by

decreasing the panel density $\bar{m}$ and size or increasing the bending stiffness:

$$D = \frac{Et^3}{12(1-\nu^2)} \quad [\text{Nm}] \quad (3.3)$$

where E = Young's modulus $\left(\frac{\text{N}}{\text{m}^2}\right)$

t = thickness (m)

ν = Poisson's ratio

3.2 Experimental Results

Tests have been carried out on aluminum, plexiglass and steel panels of various thicknesses. A honeycomb panel and aluminum panels with two and four stiffeners have also been tested. The transmission loss curves of these different panels can be found in Chapter 7.

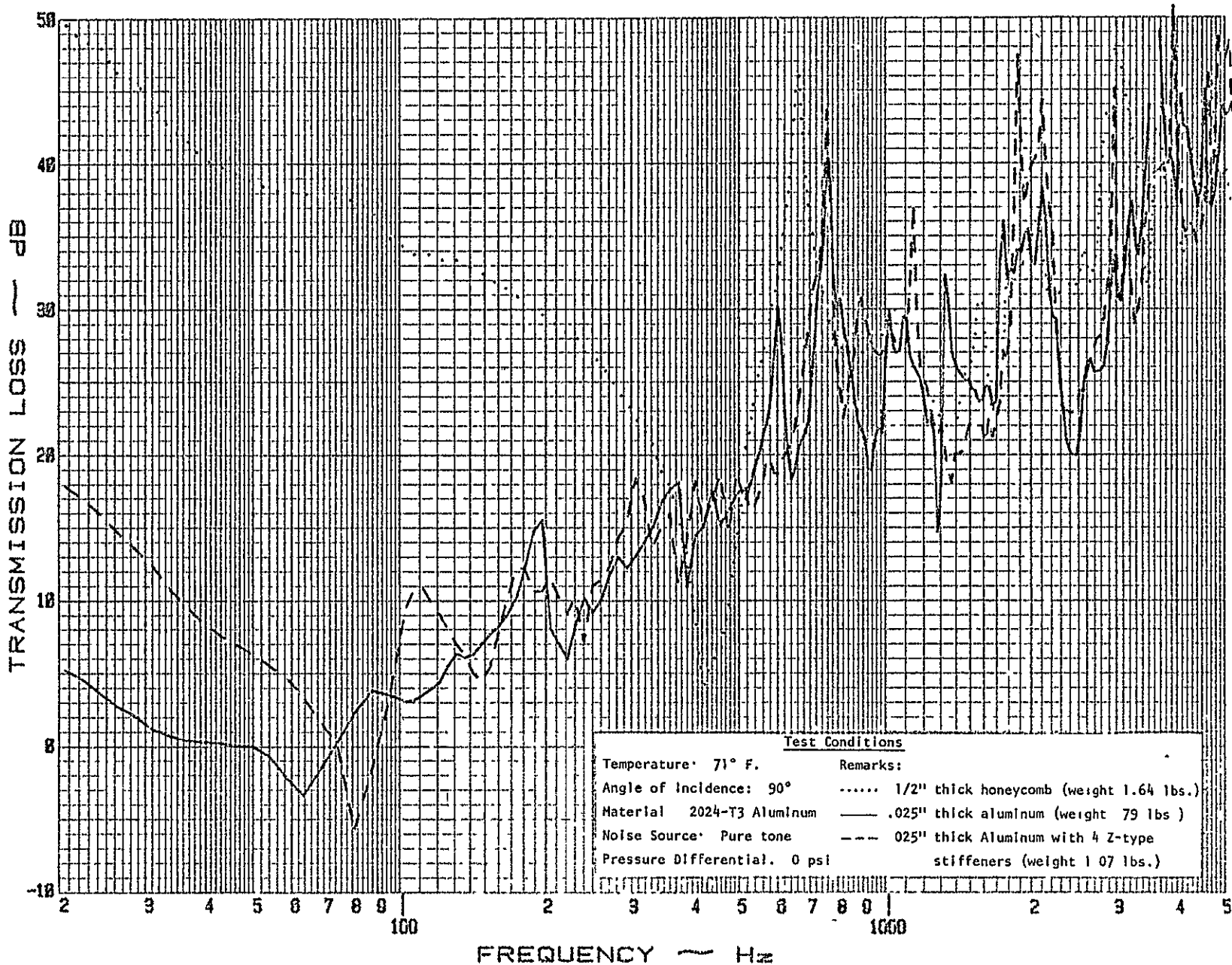
In Figure 3.1 it is shown how the bending stiffness influences the transmission characteristics. If the bending stiffness of a panel is increased, the transmission loss also increases. Note especially the high transmission loss of the honeycomb panel at lower frequencies.

The fundamental resonance frequencies of the test panels can easily be located as the point of lowest transmission loss.

3.3 Comparison

Figure 3.2 compares the actual transmission loss and the transmission loss predicted by Equation 3.2, in the stiffness controlled region. The predicted values are in fair agreement with the experimental results.

The experimental resonance frequencies differ slightly from the predicted frequencies. The discrepancies vary with the exact edge clamping conditions, as described in Reference 3.



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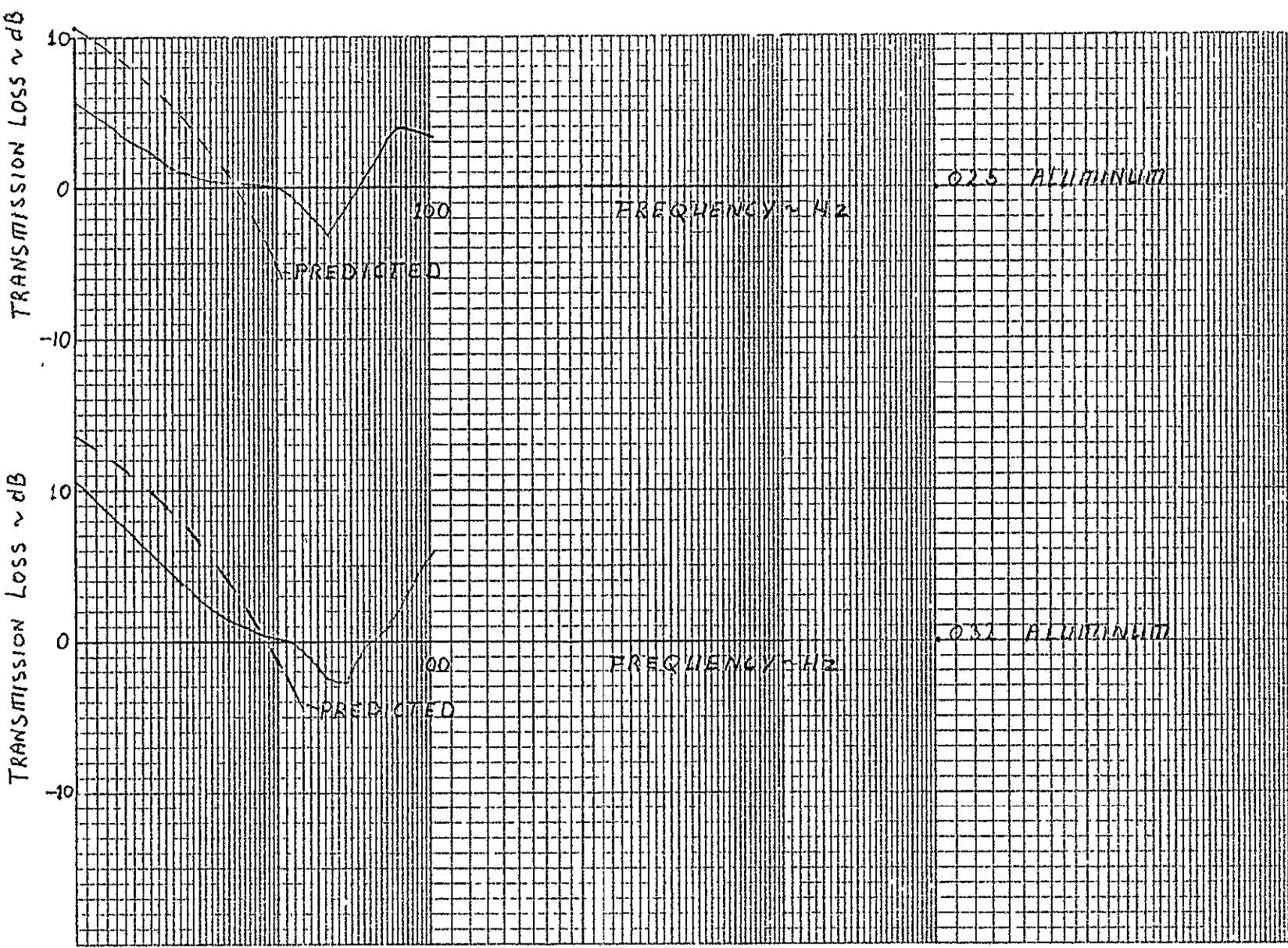


Figure 3.2: Panel behavior in stiffness controlled frequency region.

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4. The Influence of Mass

The mass of a panel has a significant influence on its vibrations and transmission loss. The influence of mass will be shown by analytical equations and by experimental results.

4.1 Analytical Considerations

Above the frequency range of the lower resonance modes, panels of finite dimensions behave approximately like infinite panels. Consequently, panel transmission loss above this range obeys the "mass law," which can be stated in theoretical form as:

$$TL = 10 \log \left[1 + \left(\frac{\omega m}{2\rho c} \cos \theta \right)^2 \right] \quad [dB] \quad (4.1)$$

where θ = angle of incidence of the sound (incidence normal to the panel is defined as zero degrees)

$$\omega = 2\pi f \quad \left[\frac{\text{rad}}{\text{s}} \right]$$

$$\rho = \text{density of air (kg/m}^3\text{)}$$

$$c = \text{speed of sound (m/s)}$$

This expression can be found in Reference 4. If the surface mass is doubled, the transmission loss increases by 6 dB. Equation 4.1 and other similar equations show that the transmission loss in this region is independent of stiffness and damping. From this equation it also follows that the transmission loss increases by 6 dB for each doubling of frequency.

4.2 Experimental Results

Chapter 7 contains the experimental results obtained up to mid-September. These results include the transmission loss curves of aluminum, steel and plexiglass panels of different thickness and with certain "sound reduction" treatments.

Figures 4.1 and 4.2 show the effect of thickness on transmission loss. Increased thickness gives higher transmission losses in both the mass controlled and the stiffness controlled frequency regions. Transmission loss increases with thickness in the mass law region because of the higher surface density of the panels, while the TL increase in the stiffness region is due to the higher bending

stiffness of the thicker panels. The same shift is demonstrated when comparing the transmission loss curves of aluminum panels with different thicknesses.

4.3 Comparison

The theory predicts that the transmission loss in the mass controlled frequency region will increase 6 dB per doubling of frequency. The straight line which is drawn in the figures of Chapter 7 represents the least-squares curve fit of the data of that particular frequency region. The slope of this line, in almost all the figures, falls between 5 and 7 dB per octave, showing fair agreement between theory and experiments. When the weight is doubled, the transmission loss is predicted to increase by 6 dB. In Figures 4.1 and 4.2 there is an increase of 6 dB at low frequencies. However, the slope of the curve decreases, so the difference between the two least-squares curve fit lines is smaller than 6 dB at higher frequencies. This difference may be due to coincidence phenomena which reduce the transmission loss below mass law at frequencies near the coincidence frequency. Such an effect would indeed be expected to be stronger for thicker panels. Figures 4.3 and 4.4 compare the mass law transmission loss (Equation 4.1) with the average (least-squares curve fit) of the experimental data. The difference of 3 dB is typical for the KU-FRL test facility. Other test facilities have reported similar variations between theory and experimental results, as stated in Reference 3.

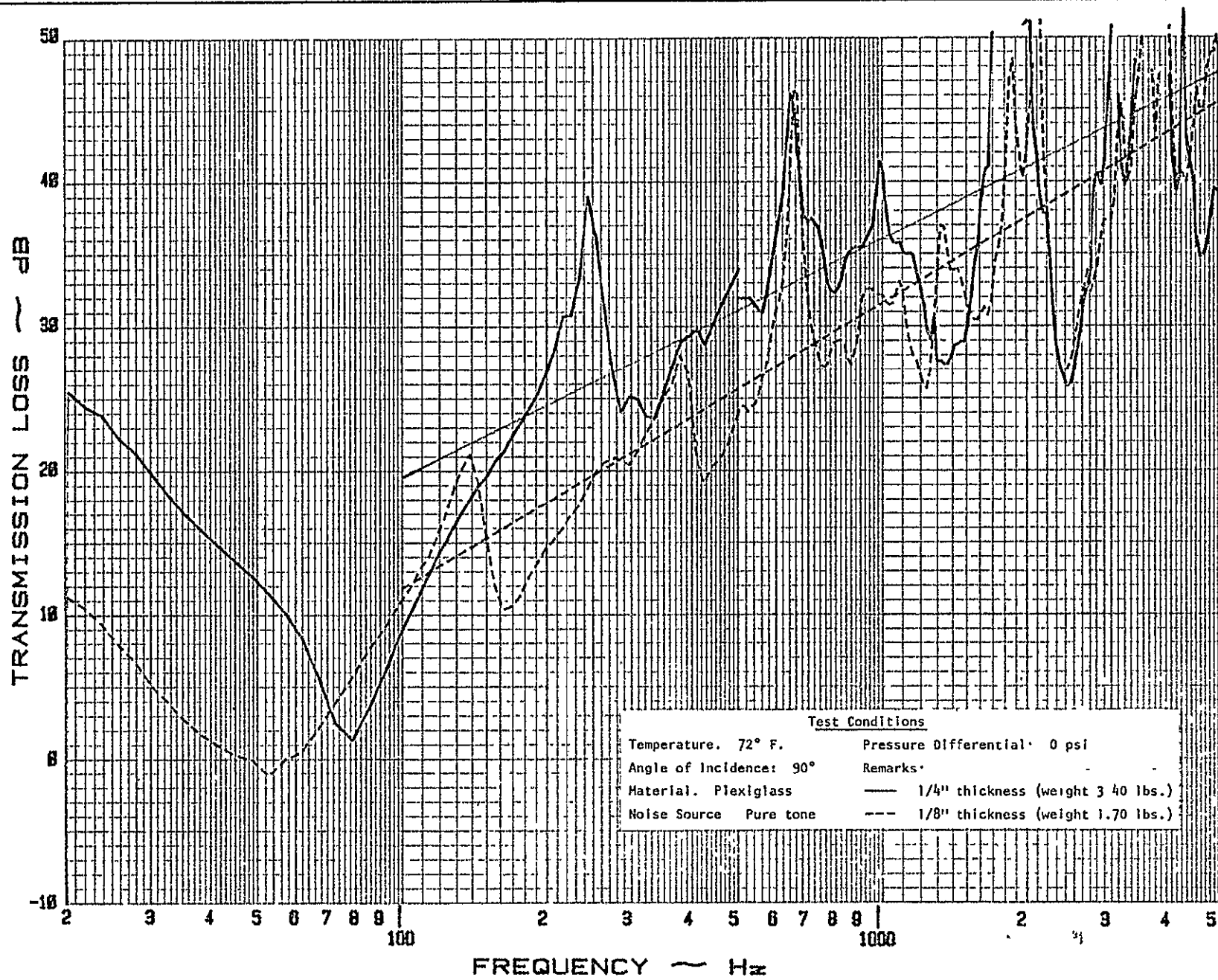


Figure 4.1: The effect of thickness on the sound transmission characteristics of an 18" x 18" panel.

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PAGE					13

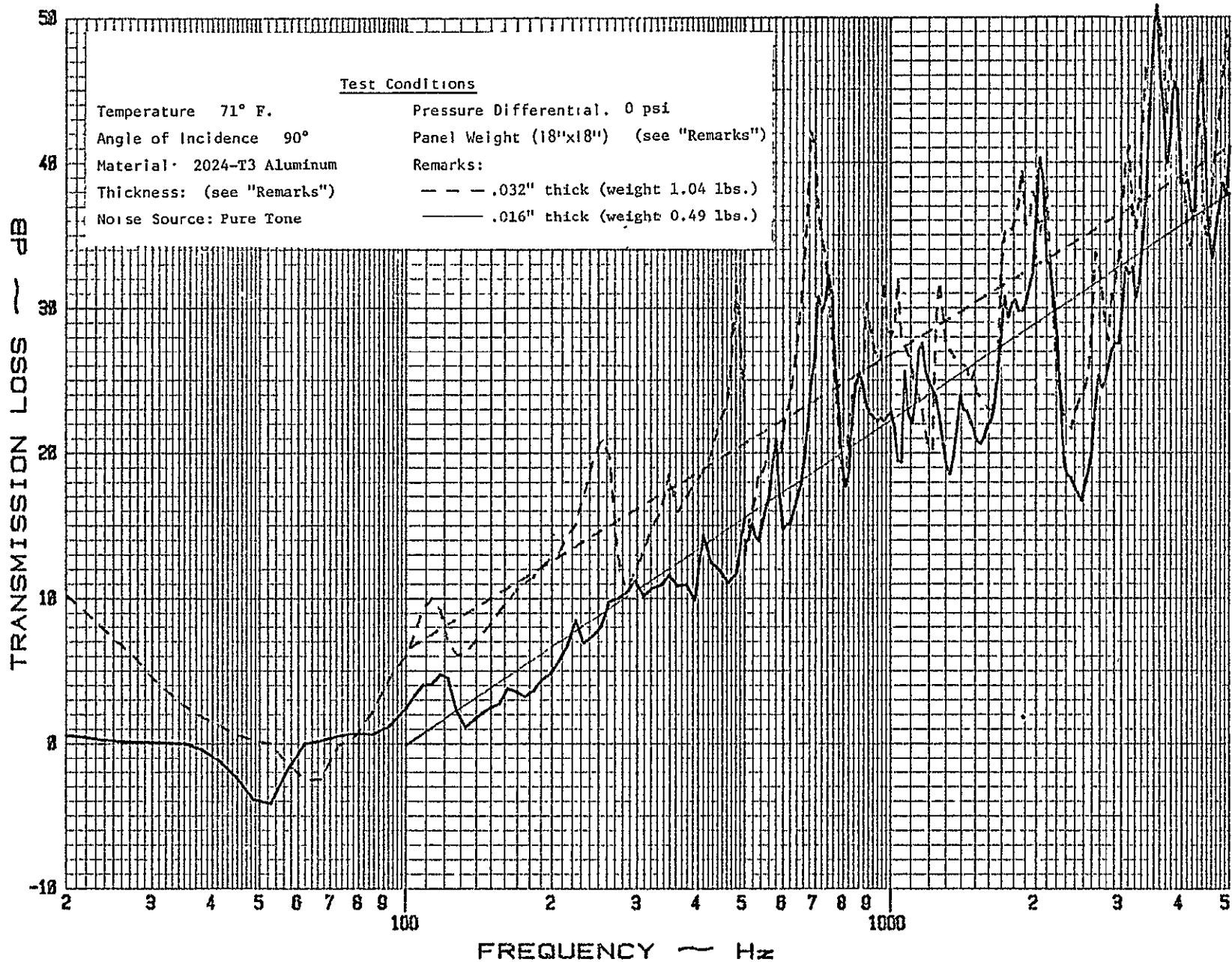
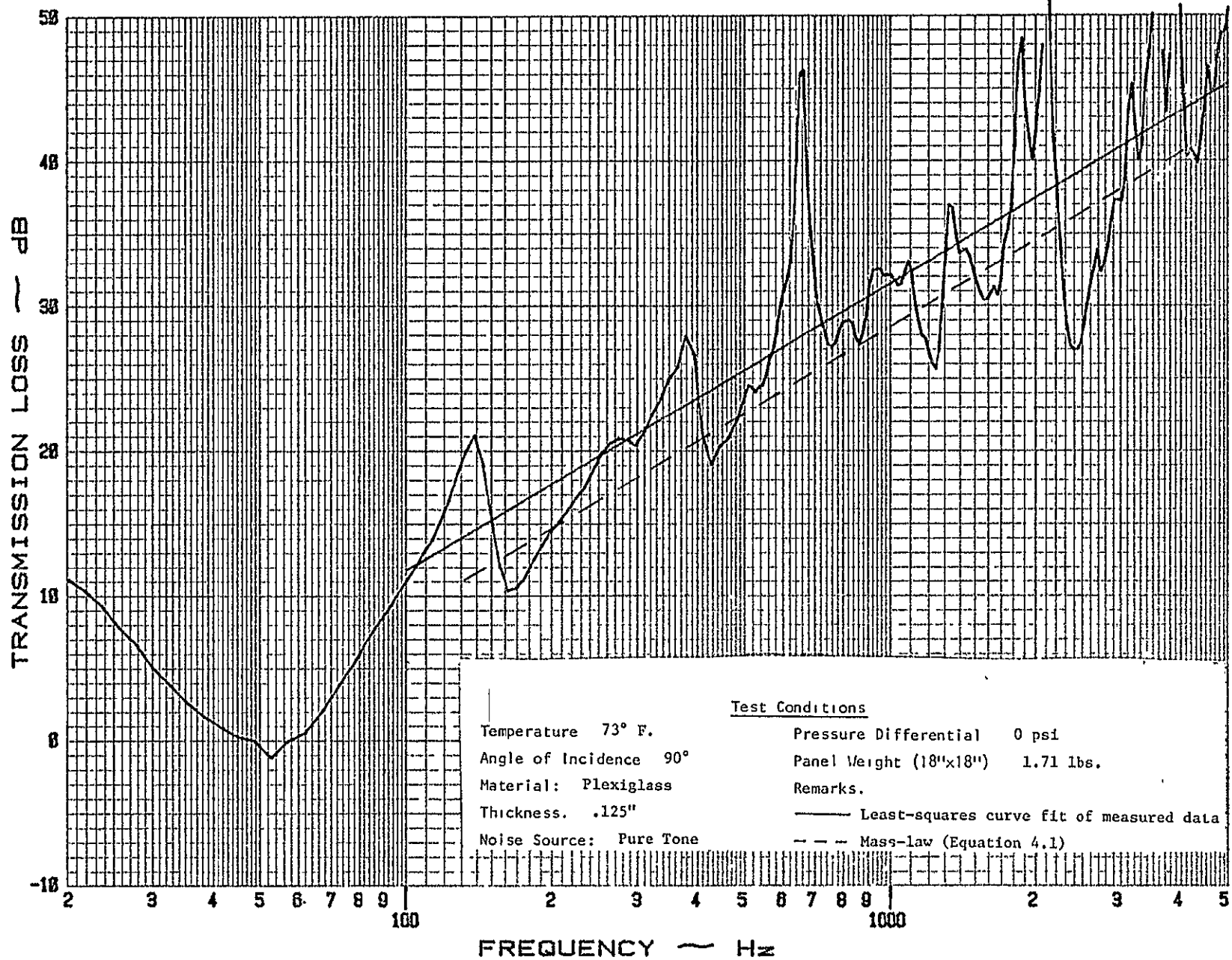


Figure 4.2: The effect of thickness on the sound transmission characteristics of 18" x 18" panels.



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Figure 4.3: Comparison between predicted and measured influence of mass on the transmission loss.					
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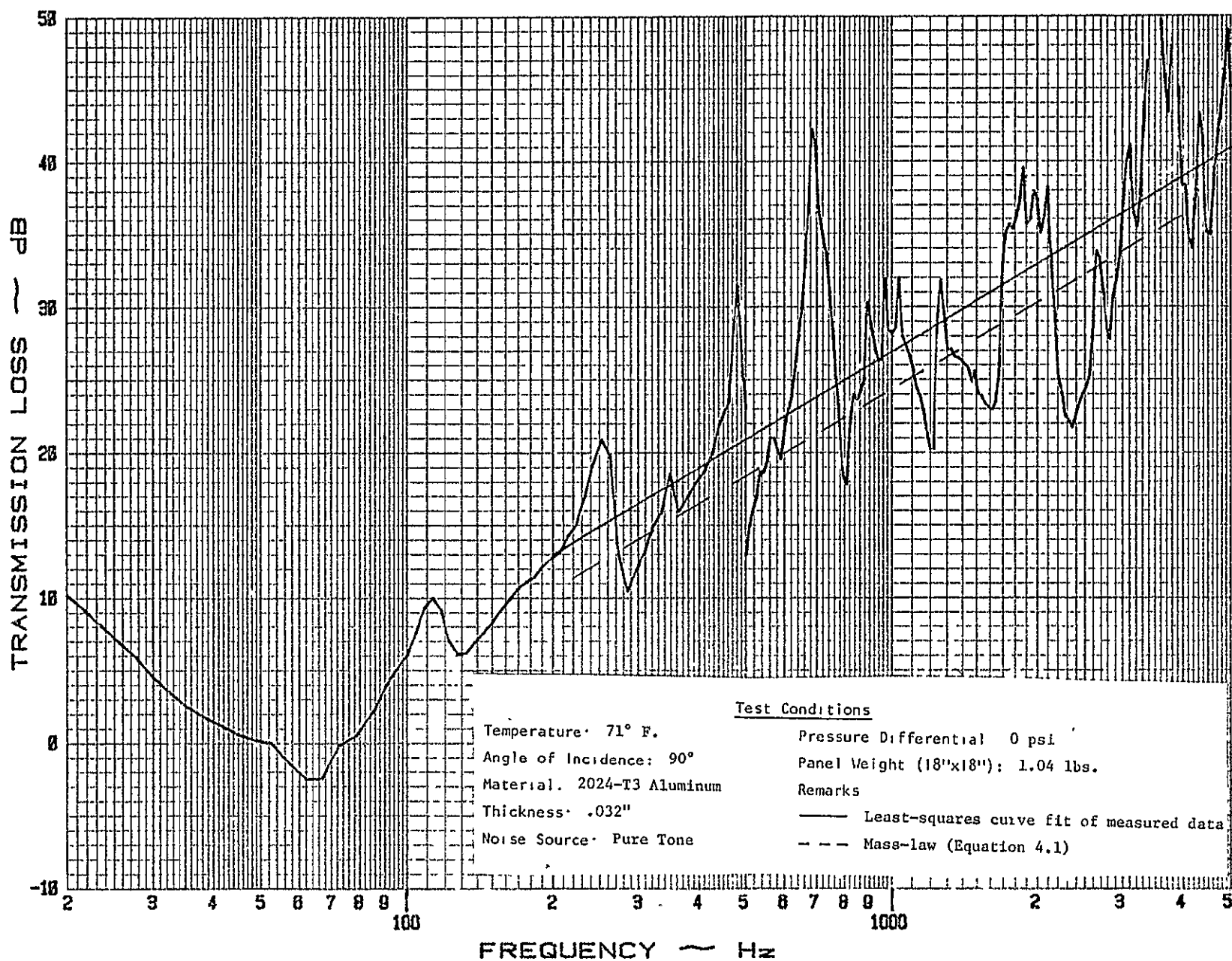


Figure 4.4: Comparison between predicted and measured influence of mass on the transmission loss.

5. The Influence of Pressurization

A pressure differential across a panel has a stiffening effect which tends to increase transmission loss below the major resonance frequency. This will be demonstrated by using an expression derived by the KU-FRL noise research team and by showing experimental results.

5.1 Analytical Considerations

Pressurization raises the transmission loss curve at lower frequencies by raising the fundamental resonance frequency several octaves. Applying a pressure differential across the panel has no effect on panel surface density, so the "mass law" transmission loss above the new fundamental resonance frequency is not changed significantly.

Several methods are available to compute the vibrations of a panel subjected to an acoustic excitation, but including the effect of a pressure differential across the panel appears more difficult. Assuming the pressure differential across the plate results in an internal membrane stress, which is constant throughout the plate, the fundamental resonance frequency of the panel is (Reference 3):

$$f_{1,1} = \frac{K}{2} \sqrt{\frac{D}{\bar{m}}} \sqrt{\pi^2 \left[\left(\frac{1}{a}\right)^2 + \left(\frac{1}{b}\right)^2 \right]^2 + \frac{N}{D} \left[\left(\frac{1}{a}\right)^2 + \left(\frac{1}{b}\right)^2 \right]} \text{ [Hz]} \quad (5.1)$$

where D = the bending stiffness of the plate (Nm)

$\bar{m}$ = the panel surface mass ($\frac{\text{kg}}{\text{m}^2}$)

a, b = the dimensions of the panel (m)

K = 1.81 for a panel with clamped edges

and the internal membrane force per unit length is:

$$N = 0.21 \, t \left[E \left(\frac{\Delta P \times a}{t} \right)^2 \right]^{1/3} \left[\frac{\text{N}}{\text{m}} \right] \quad (5.2)$$

where ΔP = pressure difference across panel ($\frac{\text{N}}{\text{m}^2}$)

Using Equation 3.2 the transmission loss for frequencies below major resonance can be calculated.

5.2 Experimental Results

To investigate the influence of pressurization the KU-FRL noise research team measured the transmission loss of several panels with

pressure differences across them. The experimental transmission loss curves of several panels at different pressure differentials can be found in Chapter 7.

Figure 5.1 shows the experimental influence of a pressure differential across an aluminum panel. This figure indicates that the first 0.5 psi of pressure differential greatly increases transmission loss at the lower frequencies. The transmission loss in the mass controlled region changes hardly at all. The same result can be seen in Figures 5.2 and 5.3.

5.3 Comparison

Figure 5.4 compares measured and predicted influence of pressure on the fundamental resonance frequency. The predicted frequencies were calculated by Equation 5.1. This figure demonstrates a fair agreement between both predicted and measured data. The comparison between measured and predicted influence of pressure on the transmission loss is shown in Figure 5.5. Here also both lines show fair agreement. Notice the large transmission loss increase resulting from a small pressure difference.

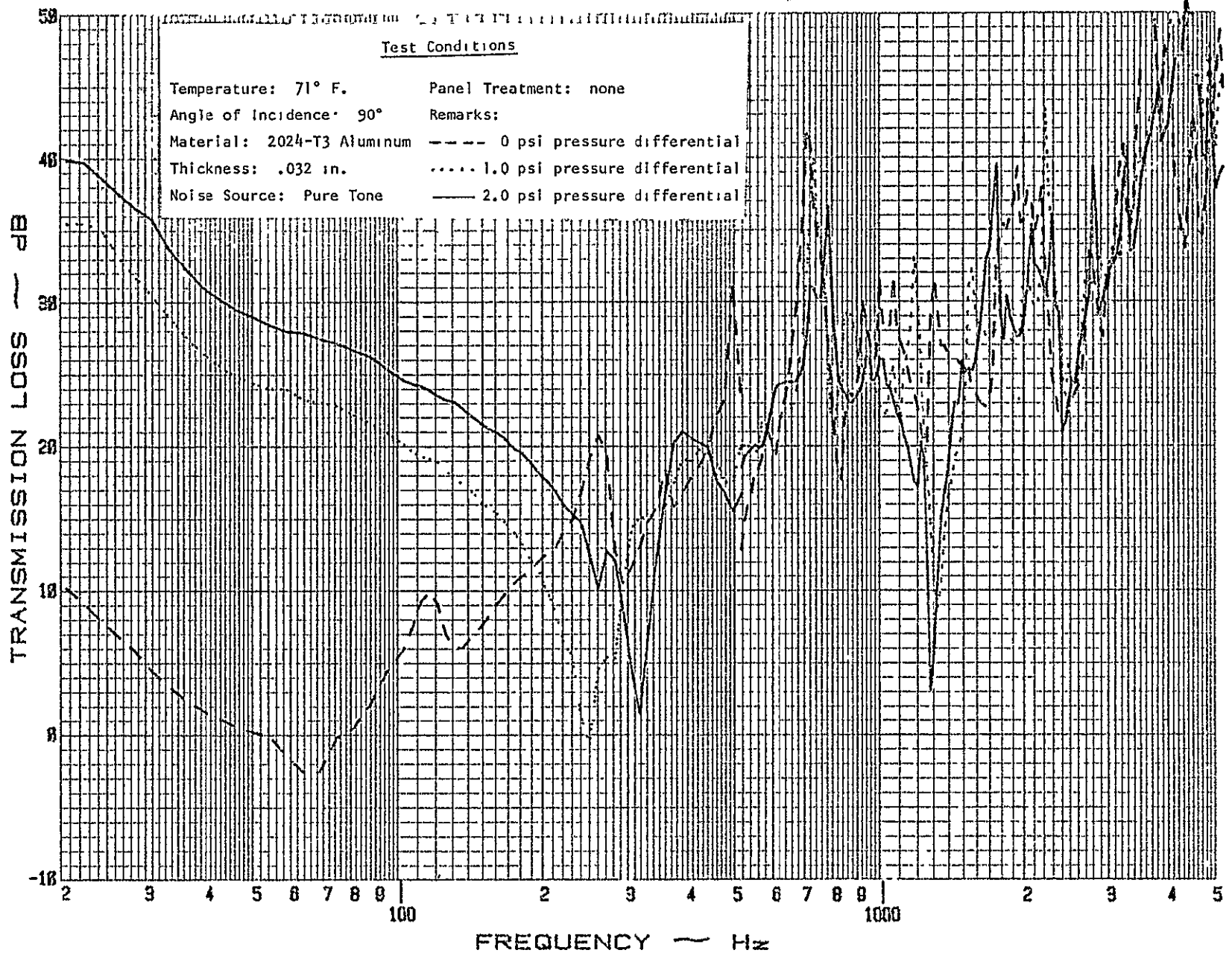


Figure 5.1: The influence of a pressure differential across an 18" x 18" panel on its sound transmission characteristics.

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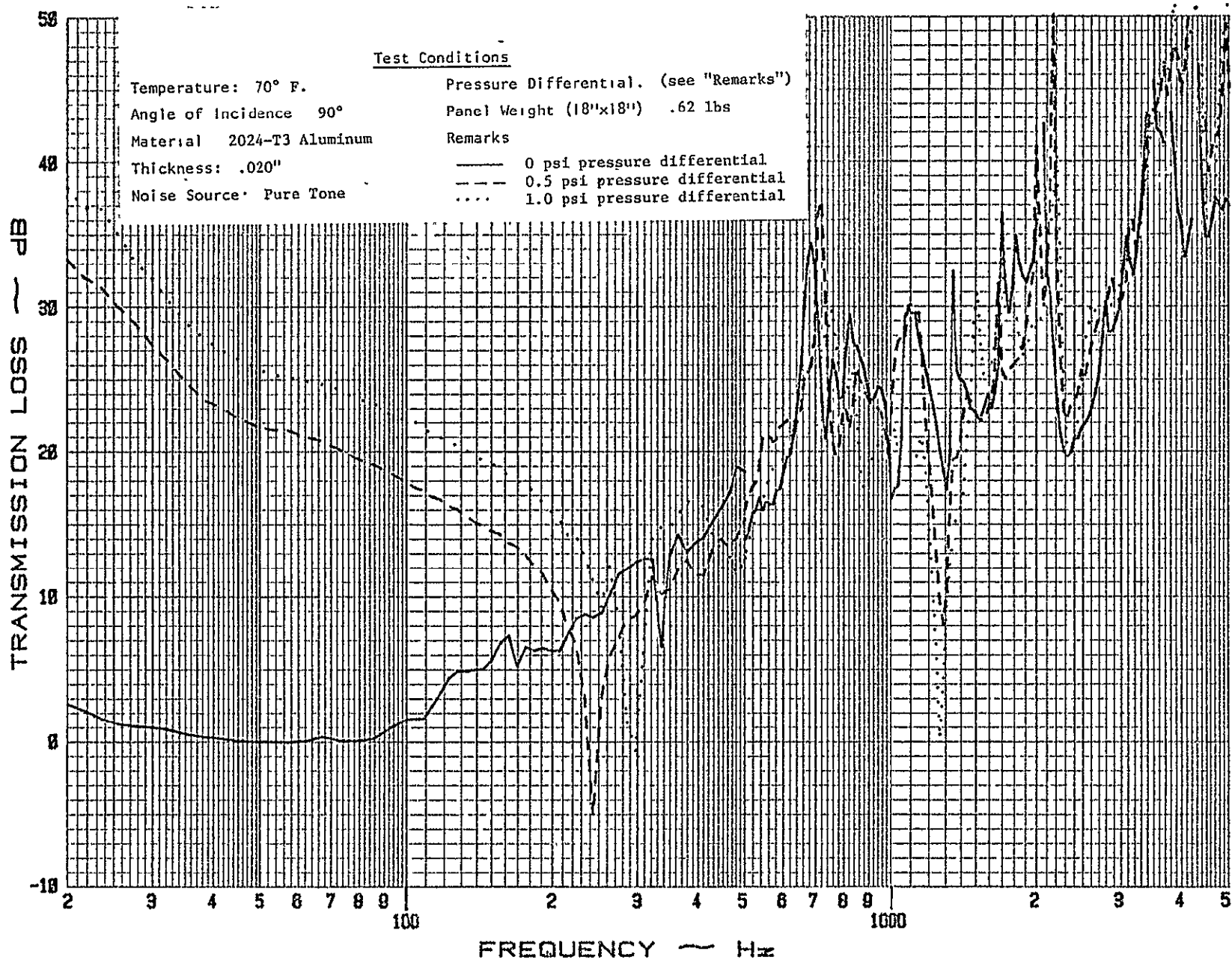
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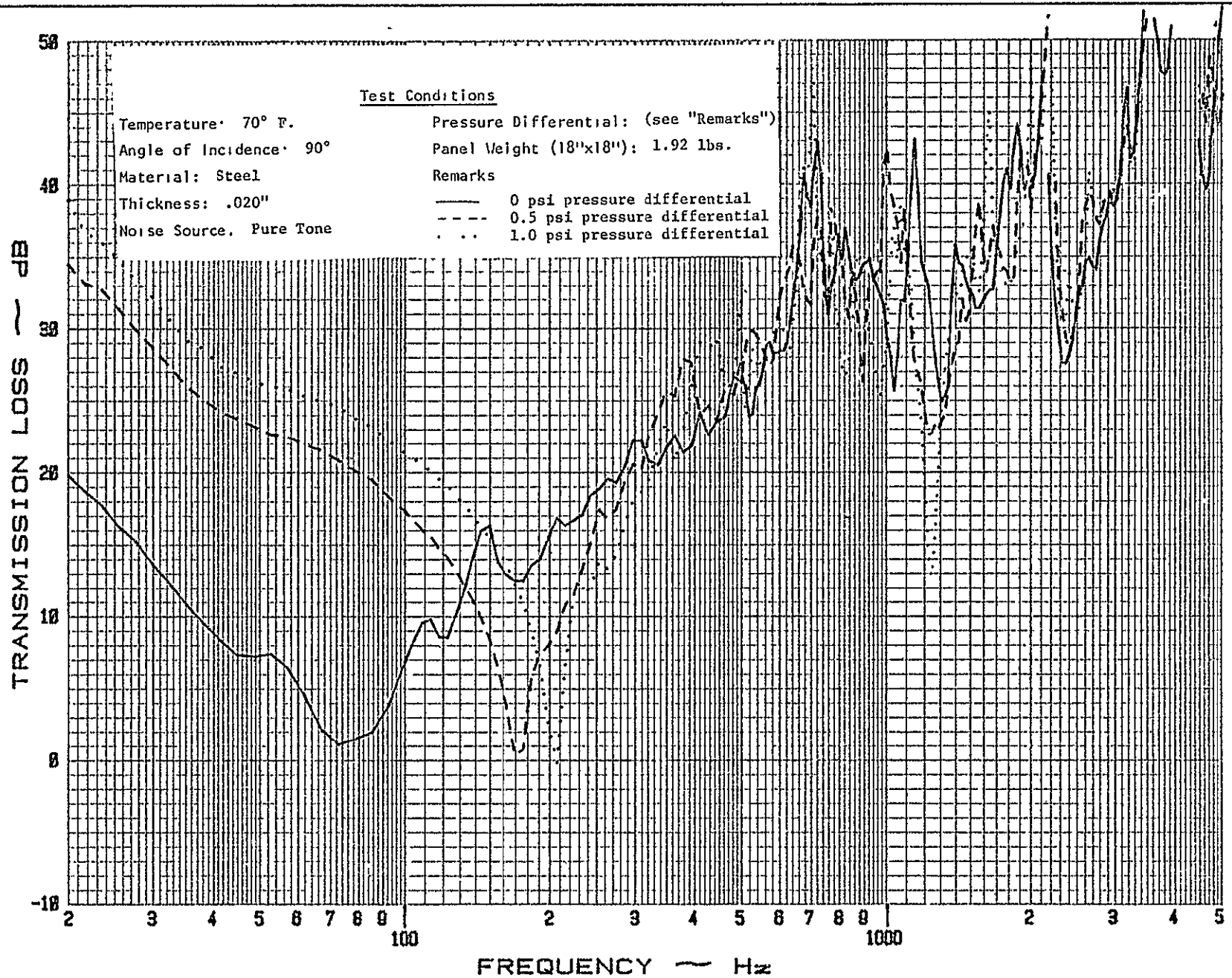
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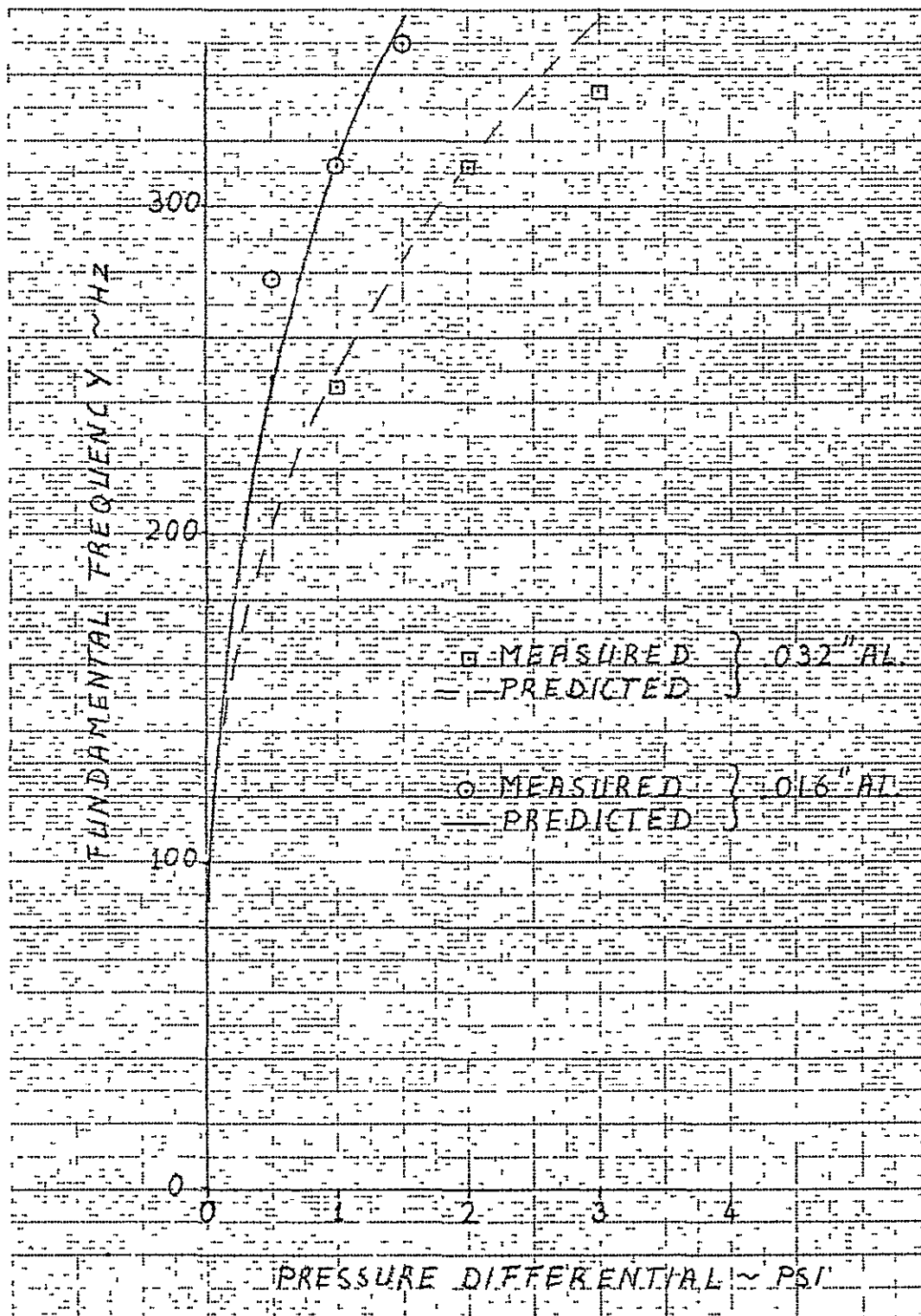
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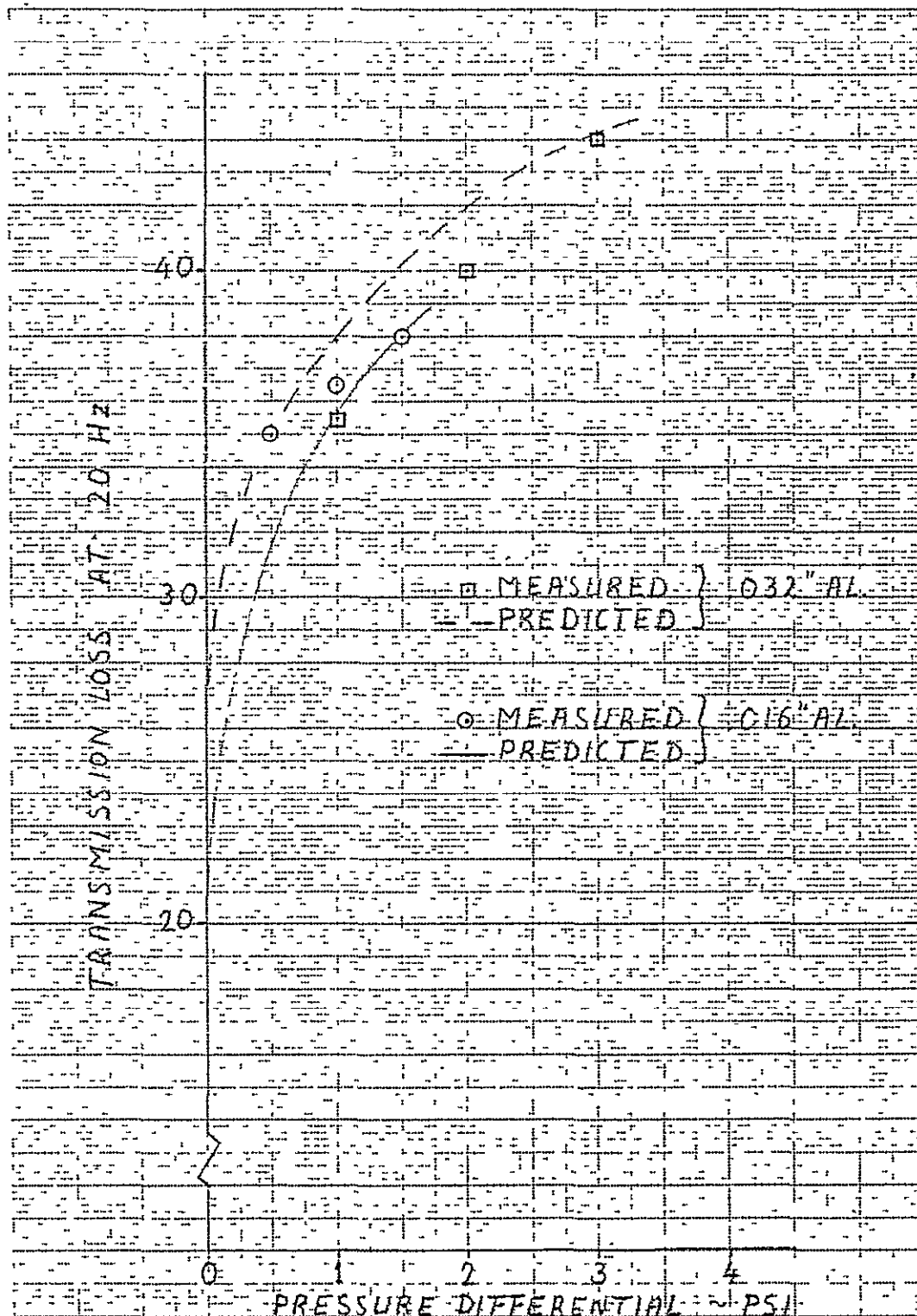
CALC			REVISED	DATE	Figure 5.2: The influence of a pressure differential across an 18" x 18" panel on the sound transmission characteristics.	
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UNIVERSITY OF KANSAS						PAGE 20







CALC			REVISED	DATE	Figure 5.4: Comparison between measured and predicted influence of pressure on the fundamental resonance frequency.	
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APPD						
APPD						
					UNIVERSITY OF KANSAS	PAGE 22



CALC			REVISED	DATE	Figure 5.5: Comparison between measured and predicted influence of pressure on the transmission loss.	
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					UNIVERSITY OF KANSAS	PAGE 23

6. The Influence of Vibration Damping Material

The dynamic responses and sound barrier properties of a structure are governed primarily by mass, stiffness and damping. The influence of stiffness and mass have already been discussed in Chapters 3 and 4.

Damping is the most complicated property and is difficult to predict. When a structure deforms, it dissipates some energy. It is this energy dissipation--this conversion of mechanical energy to thermal energy--that is called damping.

The primary effect of increased panel damping, under the conditions of KU-FRL noise research, is the reduction of vibration amplitudes at resonances. According to Reference 4, care must be exercised in damping measurements to prevent extraneous energy dissipation. To minimize support-related damping in damping measurements, it is usually necessary to suspend the test panel from two long strings. In the KU-FRL situation, however, the edges are clamped. Thus, the prediction of the total damping experienced by a test panel will be very difficult.

However, by comparing the transmission loss of a bare panel with the transmission loss of a panel treated with a viscoelastic material, the influence of the viscoelastic layer can be readily seen. To investigate possible weight savings, some panels were only partially covered with a viscoelastic layer.

In the frequency region well below the fundamental resonance, the transmission loss is independent of damping, as shown in Chapter 4. However, when a panel has been treated with a damping material, its mass will be greater, so the transmission loss in the (high frequency) mass law region will increase. Such an increase in TL is due to the greater weight of the treated panel, not to the damping material itself. The increase in transmission loss due to the increase in weight can be predicted with Equation 4.1.

6.1 Experimental Results

The KU-FRL noise research team is continuing to test the influence of vibration damping materials. Figures 6.1 shows how vibration damping materials affect the sound transmission loss of an aluminum panel. The average transmission loss is hardly changed by adding a layer of

vibration damping material; however, the resonance peaks become smaller. It can be noted that the shift of the least-squares curve fit to a higher transmission loss is due to the higher surface density of the treated panel. Figure 6.2 demonstrates the influence of partial coverage. The potential user himself must weigh the lower weight of the partially covered panel against reduced sound blockage of the resonance frequencies. Chapter 7 contains additional transmission loss curves of panels treated with vibration damping materials.

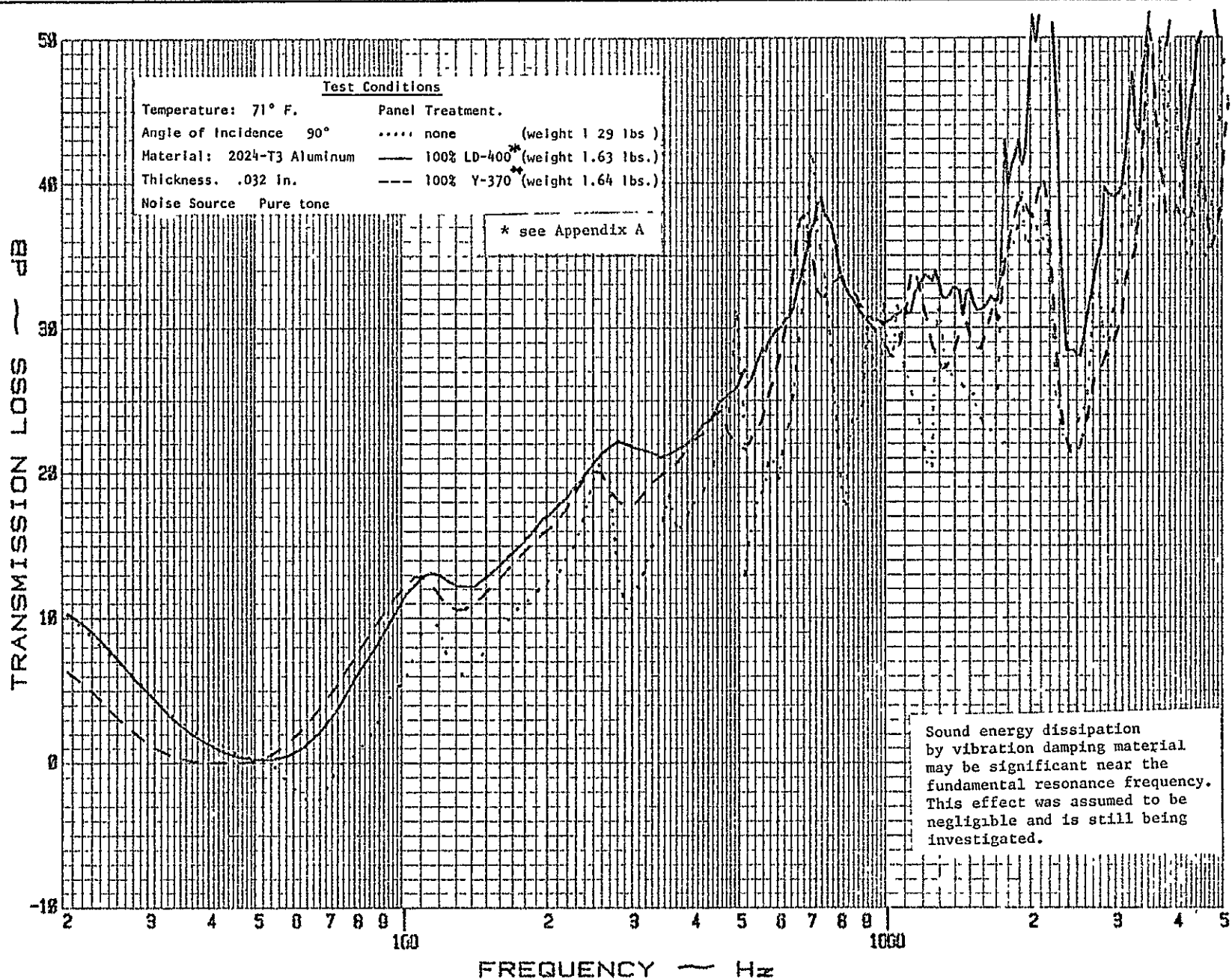


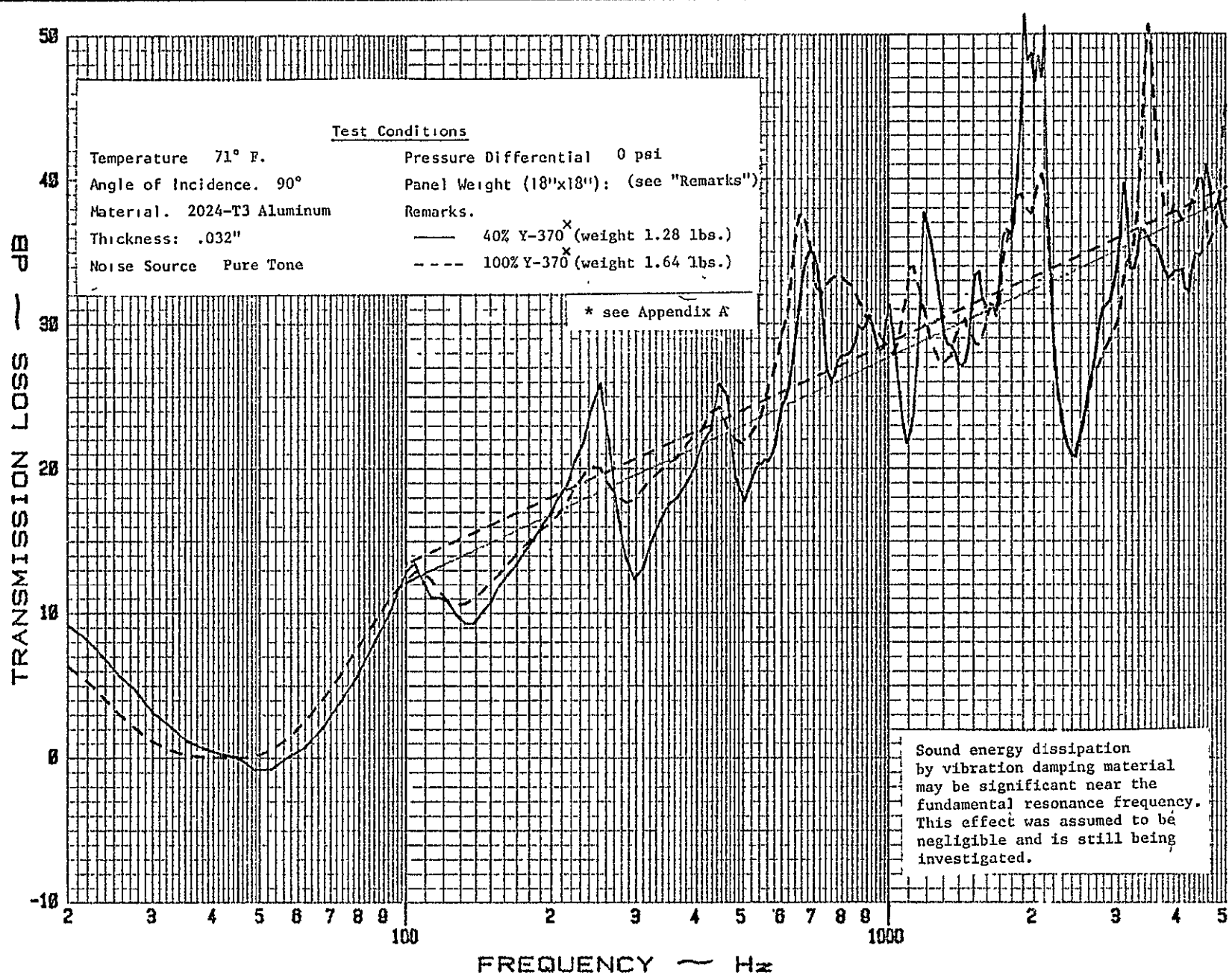
Figure 6.1: The effect of vibration damping materials on the sound transmission characteristics of 18" x 18" panels.

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Figure 6.2: The effect of vibration damping materials on the sound transmission characteristics of 18" x 18" panels.

UNIVERSITY OF KANSAS

PAGE 27



7. Test Results

The tests in the following table have been carried out up to mid-September. The figures referred to in the table can be found on the pages following. Numbers in parentheses refer to the manufacturers of the acoustic materials. See Table 7.2.

Table 7.1 List of Figures

<u>Figure</u>	<u>Material</u>	<u>Thickness (in.)</u>	<u>Test Condition or Treatment</u>
7.1	Aluminum	.016	bare
7.2	Aluminum	.020	bare
7.3	Aluminum	.025	bare
7.4	Aluminum	.032	bare
7.5	Aluminum	.040	bare
7.6	Steel	.016	bare
7.7	Steel	.020	bare
7.8	Plexiglass	.125	bare
7.9	Plexiglass	.1875	bare
7.10	Plexiglass	.250	bare
7.11	Aluminum	.032	40% Y-370 (1)
7.12	Aluminum	.032	70% Y-370 (1)
7.13	Aluminum	.032	100% Y-370 (1)
7.14	Aluminum	.032	39.5% LD-400 (2)
7.15	Aluminum	.032	63.2% LD-400 (2)
7.16	Aluminum	.032	92.2% LD-400 (2)
7.17	Aluminum	.032	100% LD-400 (2)
7.18	2 x Aluminum	.016	IC-998 (3) between sheets
7.19	Aluminum	.025	2 stiffeners
7.20	Aluminum	.025	4 stiffeners
7.21	Aluminum	.025	4 stiffeners + 1.0 psi
7.22	Aluminum	.025	4 stiffeners + 2.0 psi
7.23	Aluminum	.025	4 stiffeners + 3.0 psi
7.24	Aluminum	.025	4 stiffeners + Y-370 in the middle
7.25	Aluminum	.025	4 stiffeners + Y-370 in the middle + 1.0 psi

Table 7.1 List of Figures (continued)

<u>Figure</u>	<u>Material</u>	<u>Thickness (in.)</u>	<u>Test Condition or Treatment</u>
7.26	Steel	.020	0.5 psi
7.27	Steel	.020	1.0 psi
7.28	Steel	.020	2.0 psi
7.29	Aluminum	.016	0.5 psi
7.30	Aluminum	.016	1.0 psi
7.31	Aluminum	.016	1.5 psi
7.32	Aluminum	.020	0.5 psi
7.33	Aluminum	.020	1.0 psi
7.34	Aluminum	.020	1.5 psi
7.35	Aluminum	.032	1.0 psi
7.36	Aluminum	.032	2.0 psi
7.37	Aluminum	.032	3.0 psi
7.38	Honeycomb	.50	bare
7.39	Honeycomb	.50	100 % Y-370 (1)
7.40	"Noiseless Steel" (4)	≈.050	2 sheets of steel with vibration damping material between them

Table 7.2 List of Manufacturers of Acoustic

Materials

- 1 3-M, Minneapolis, Mn.
- 2 Lord Corp., Erie, Pa.
- 3 Insul-Coustic/Birma Corp.,
Sayville, N.Y.
- 4 Specialty Composites Corp.,
Newark, Del.

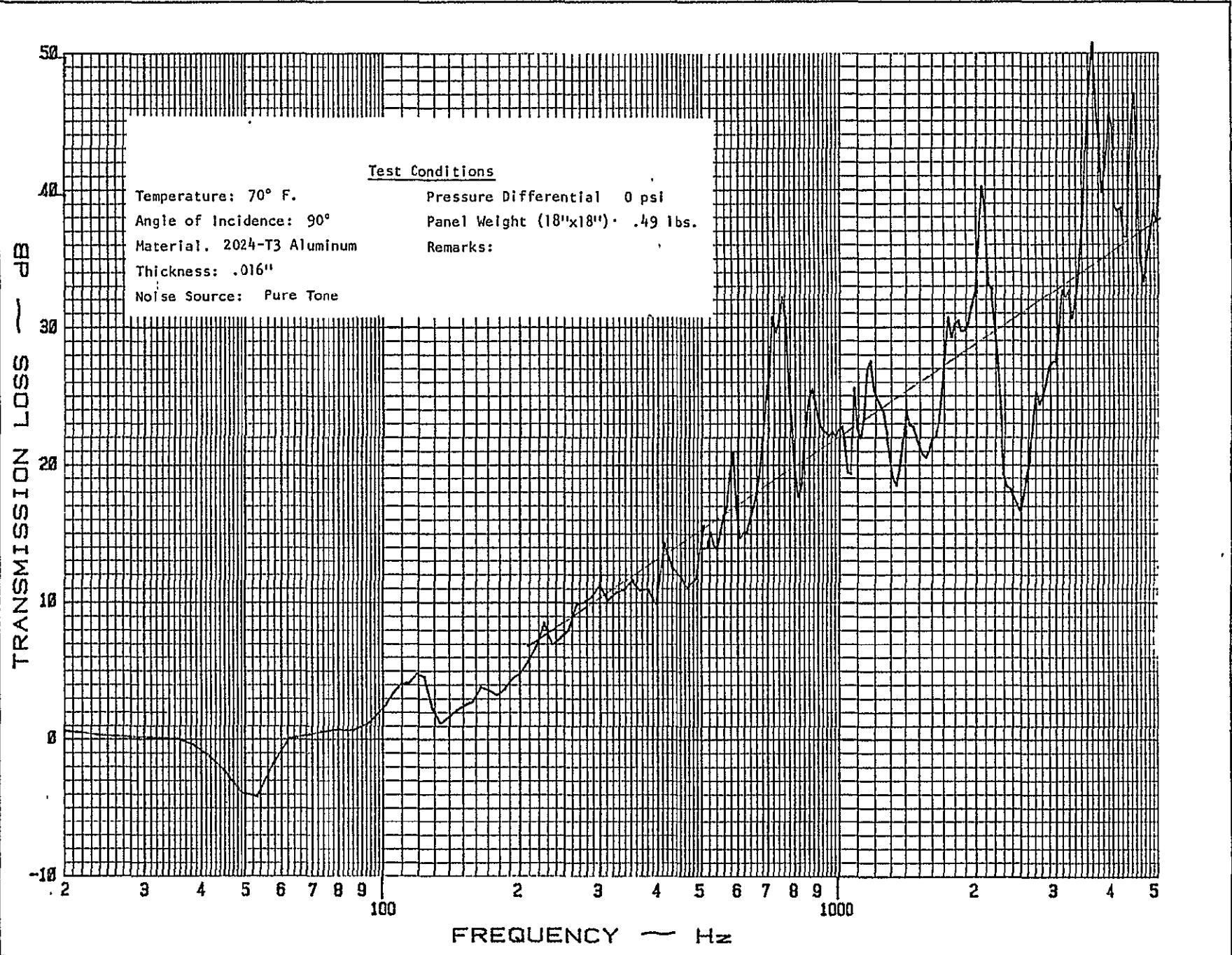
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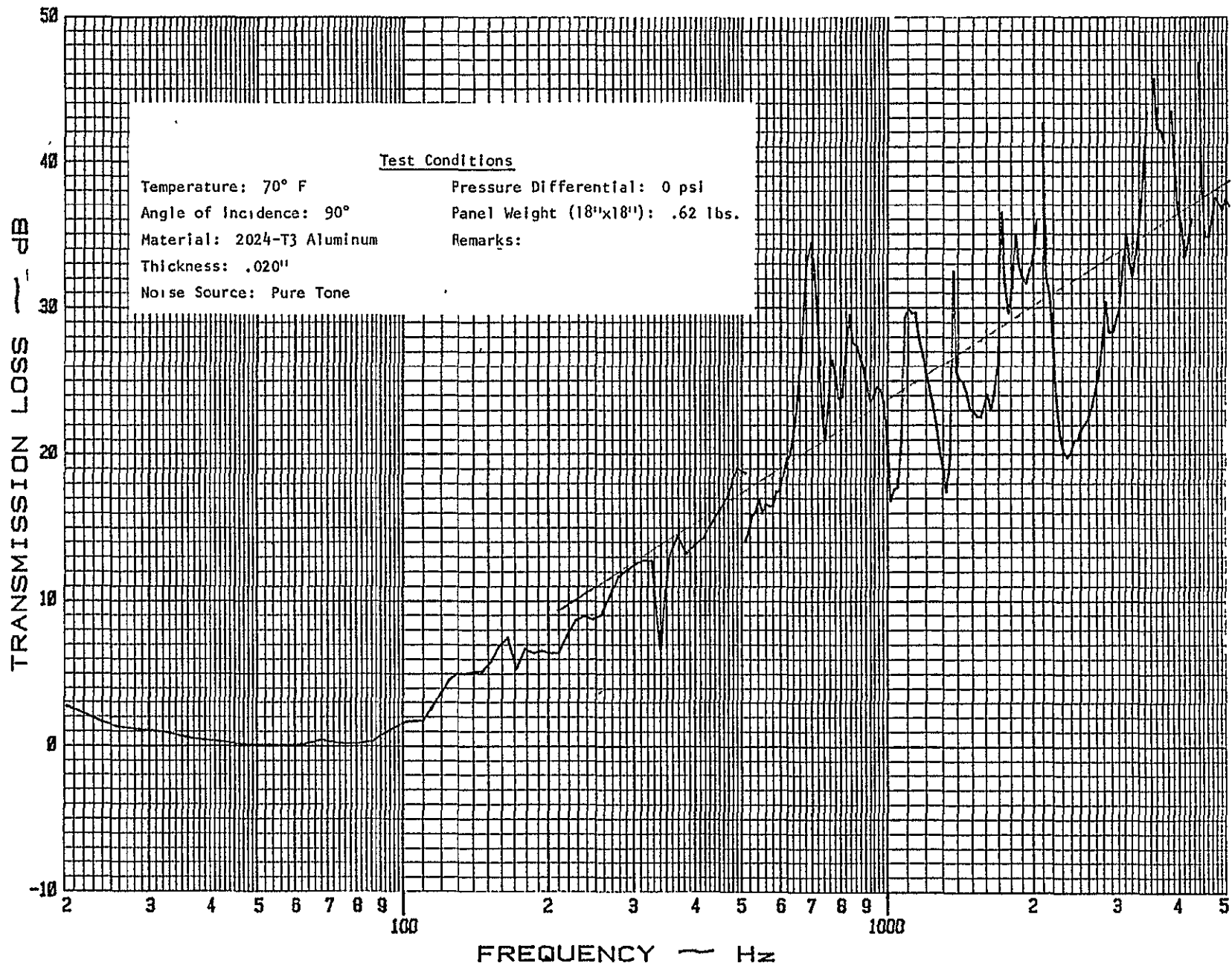
UNIVERSITY OF KANSAS

PAGE 30

Figure 7.1: Experimental sound transmission loss of an aluminum panel.

30





CALC		REVISED	DATE	UNIVERSITY OF KANSAS	PAGE 31
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APPD					
APPD					

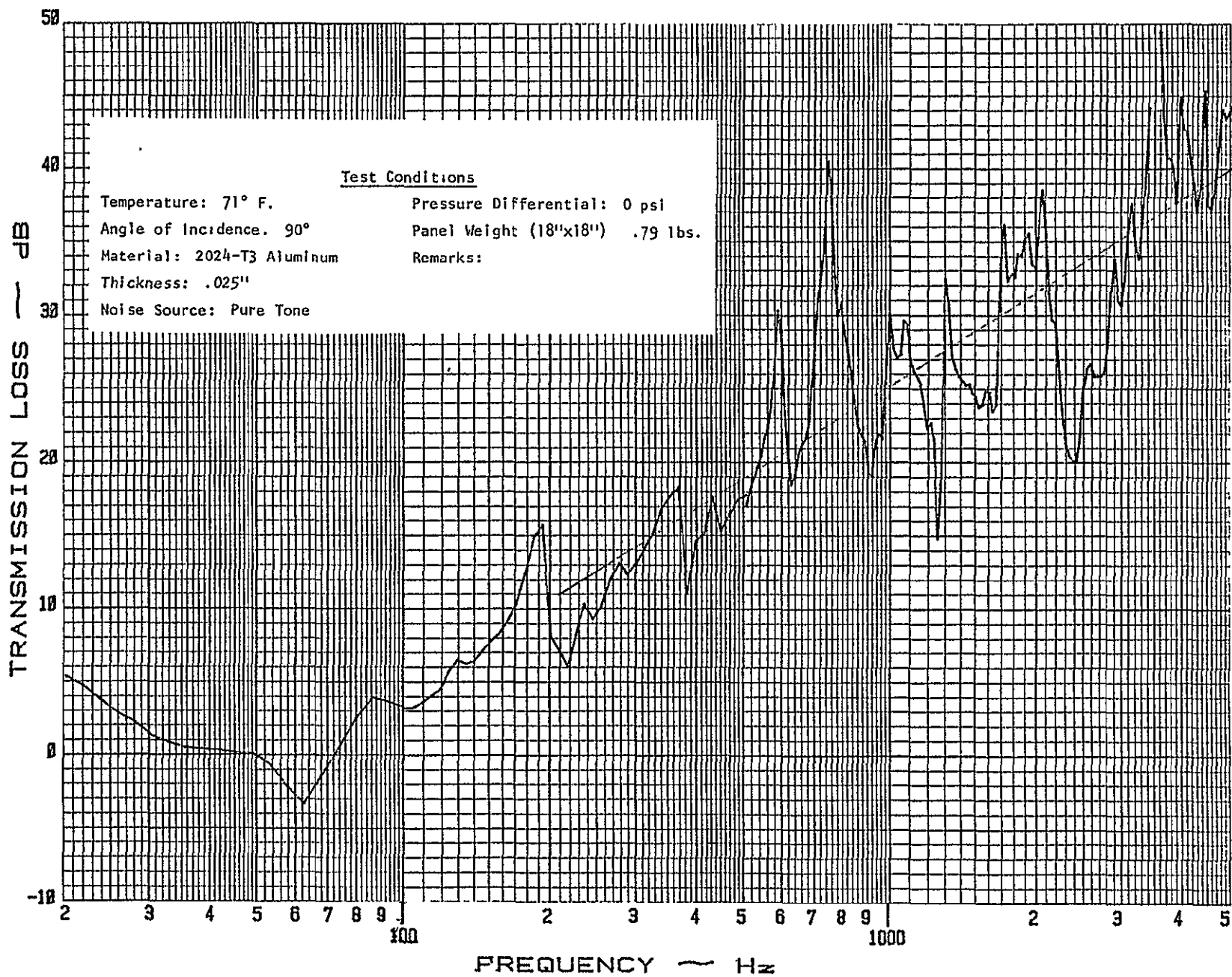


Figure 7.3: Experimental sound transmission loss of an aluminum panel.

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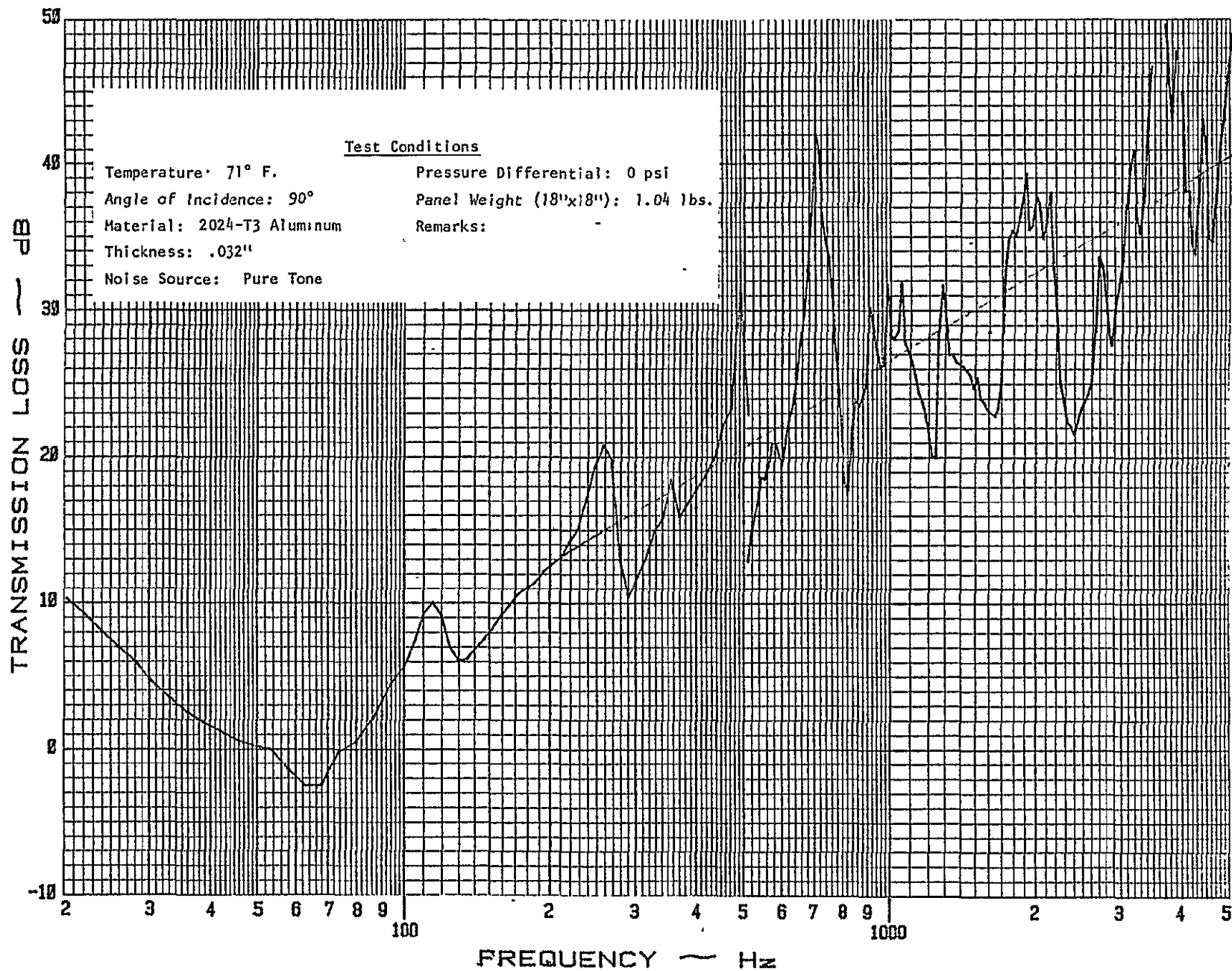
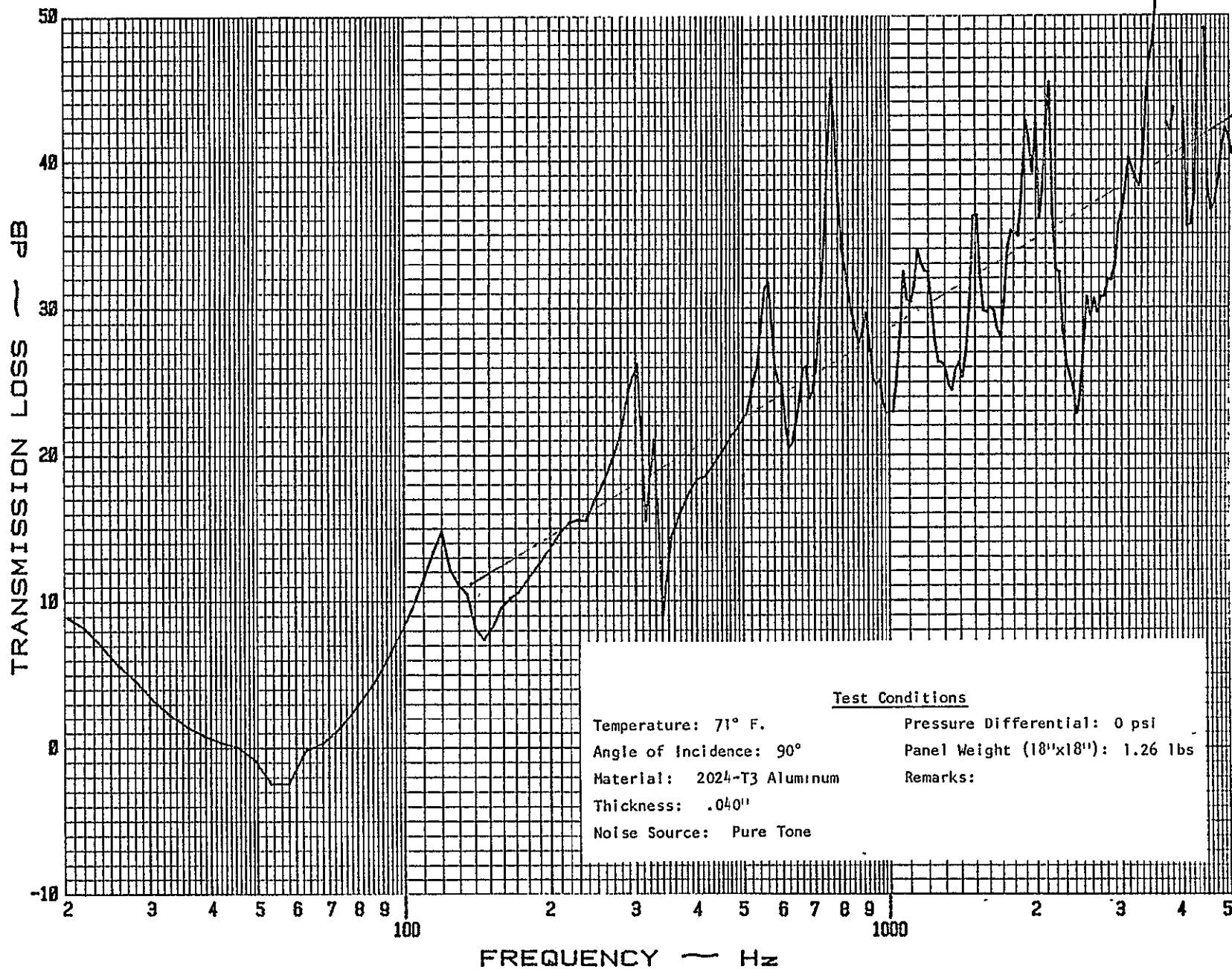


Figure 7.4: Experimental sound transmission loss of an aluminum panel.



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Figure 7.5: Experimental sound transmission loss of an aluminum panel.					

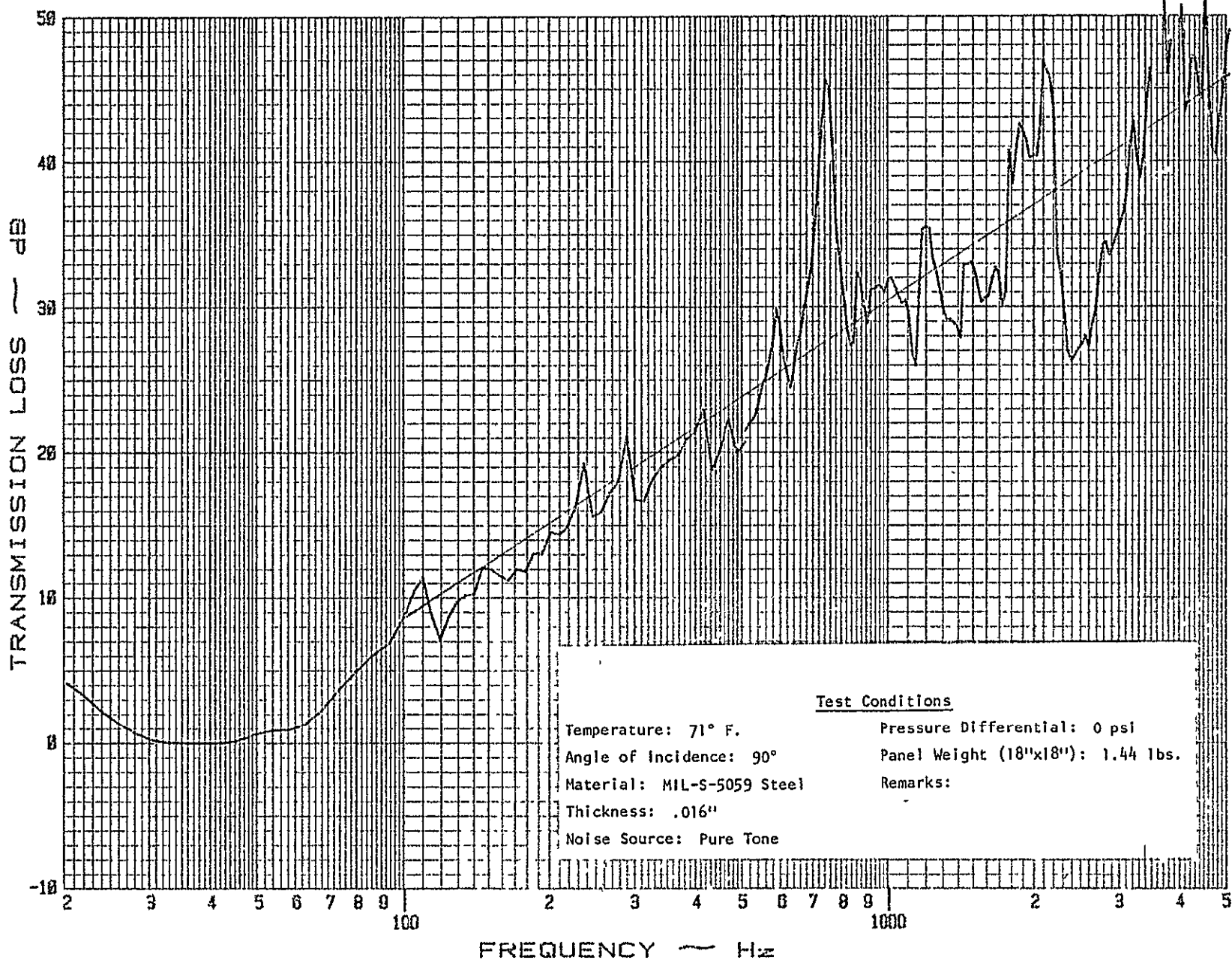


Figure 7.6: Experimental sound transmission loss of a steel panel.

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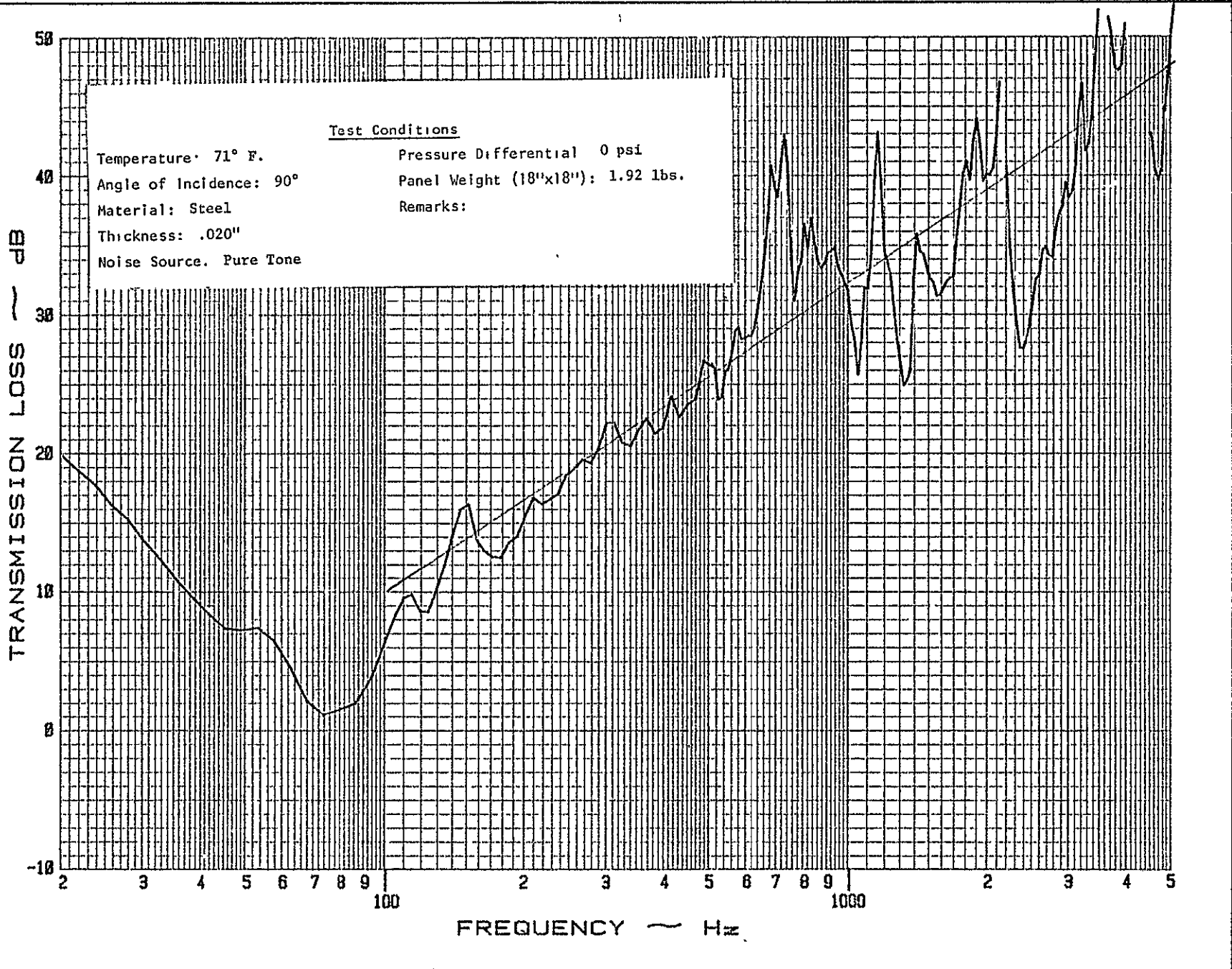
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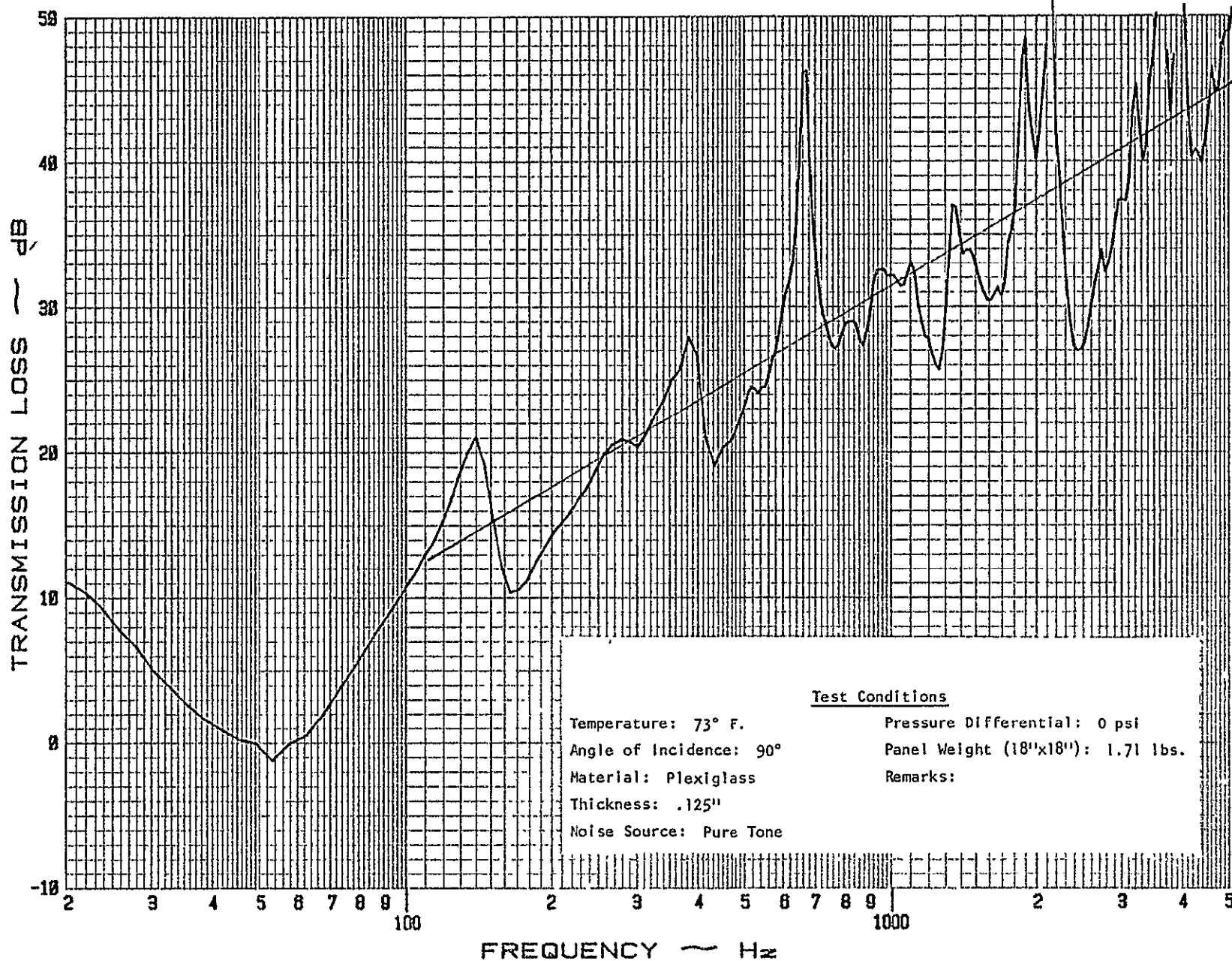


Figure 7.8: Experimental sound transmission loss of a plexiglass panel.

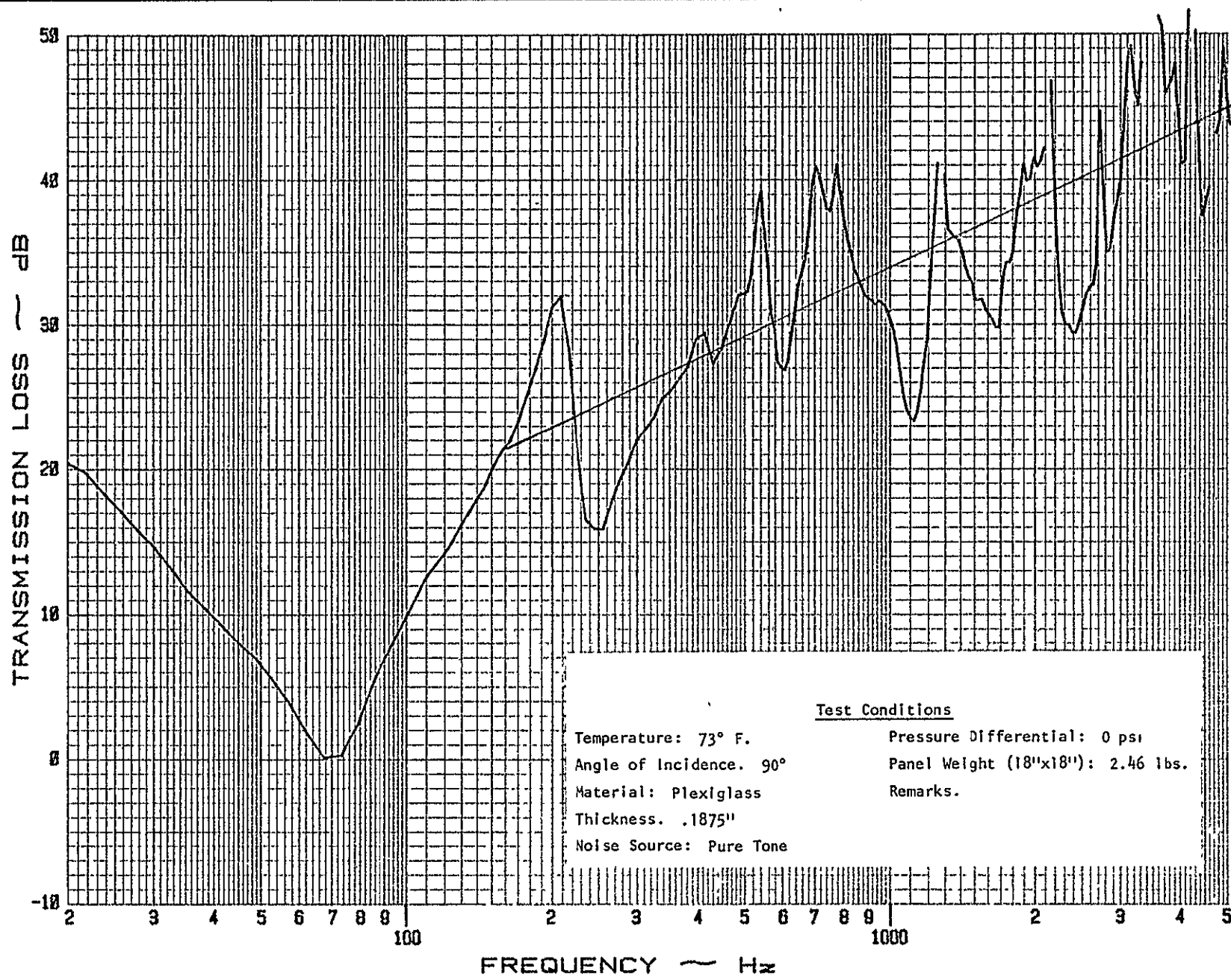


Figure 7.9: Experimental sound transmission loss of a plexiglass panel.

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UNIVERSITY OF KANSAS							PAGE 38

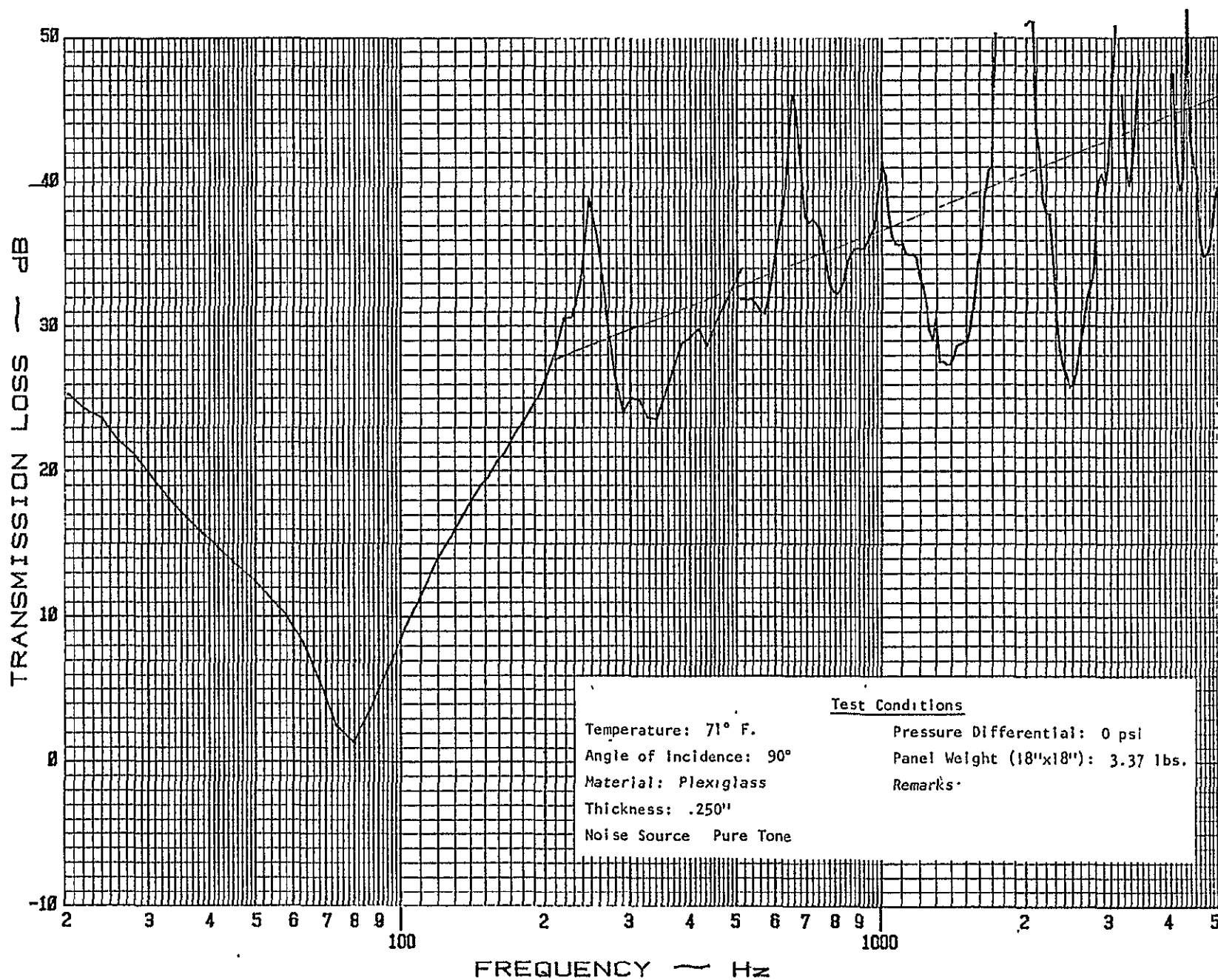
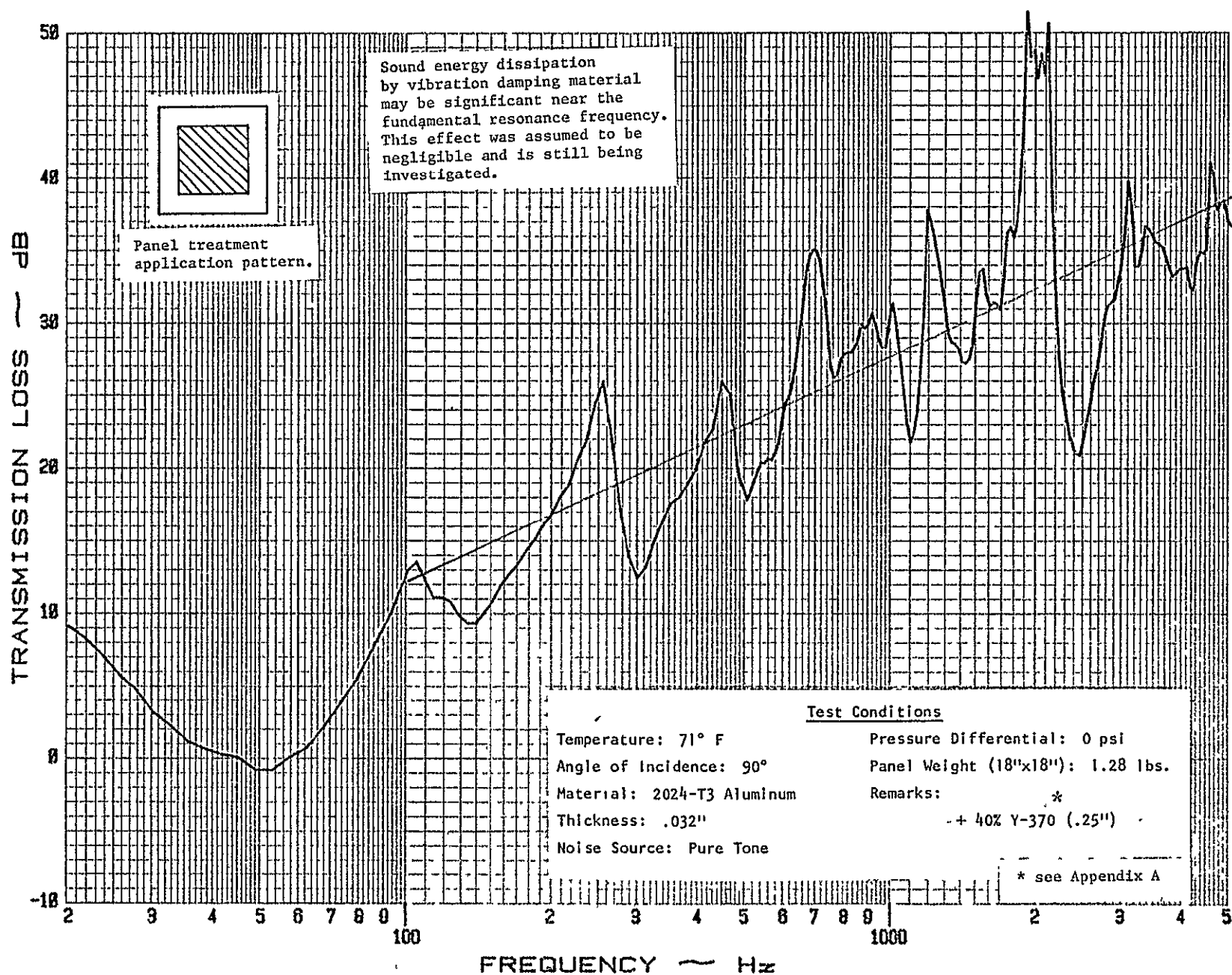


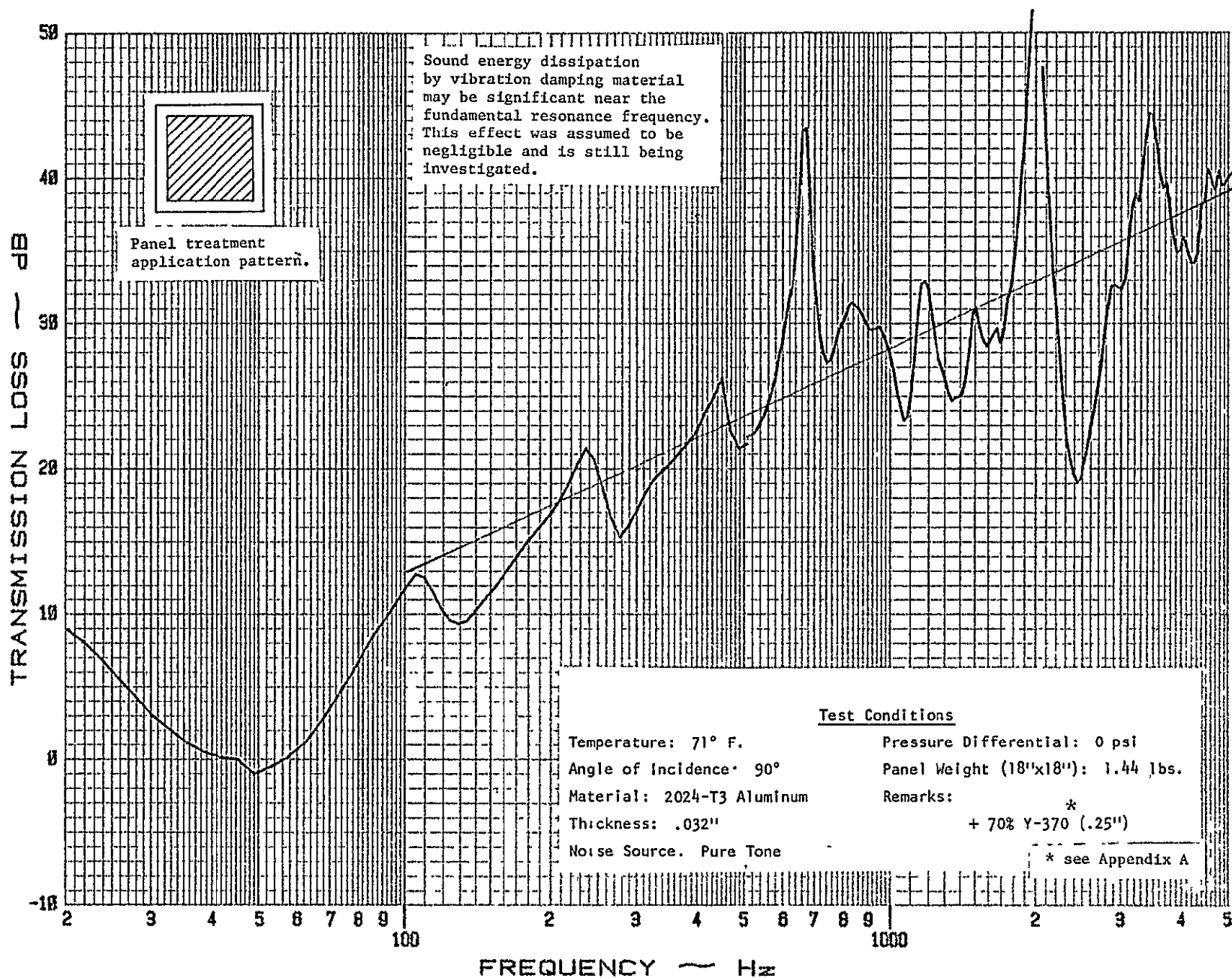
Figure 7.10: Experimental sound transmission loss of a plexiglass panel.

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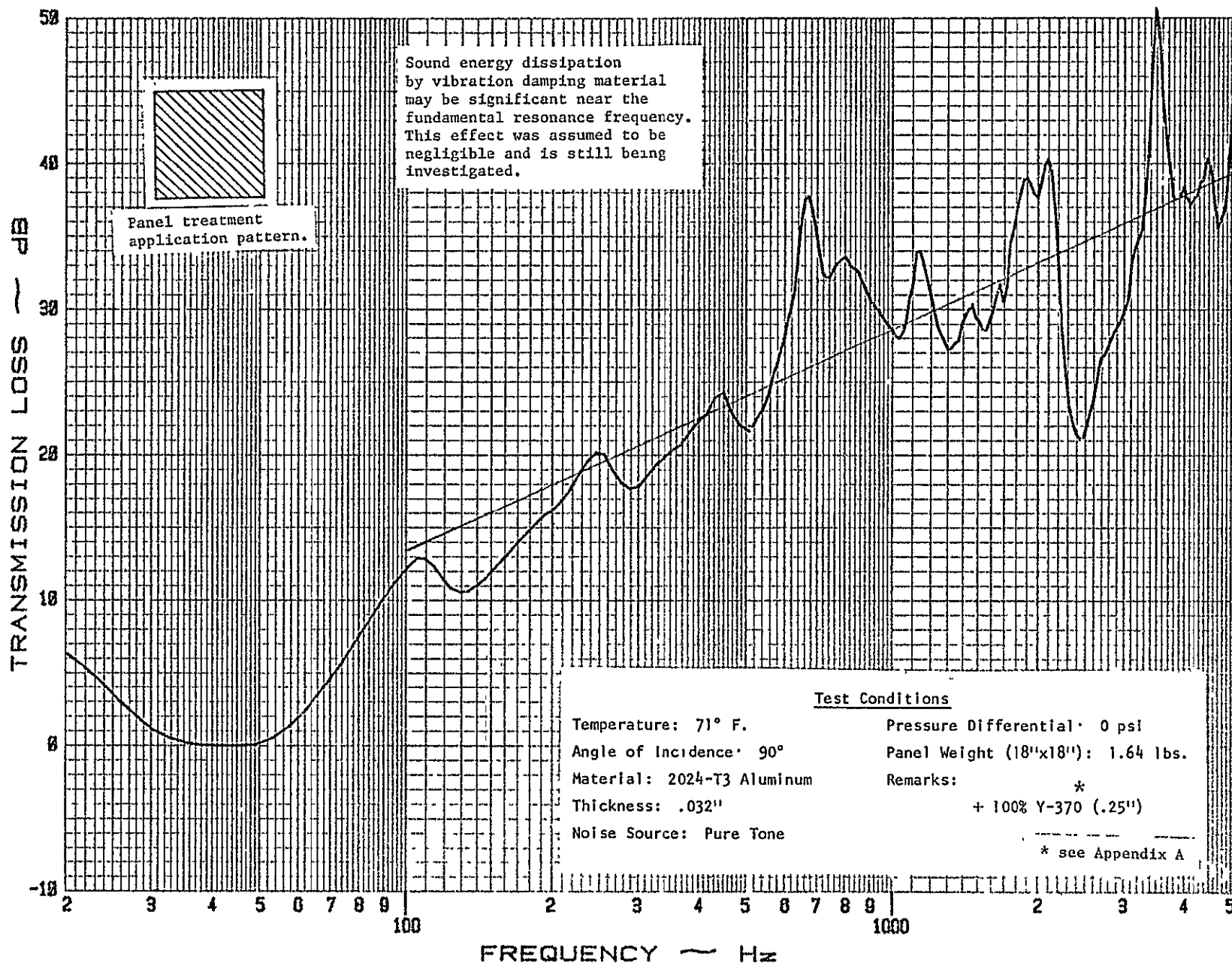
Figure 7.11: Experimental sound transmission loss of an aluminum panel.



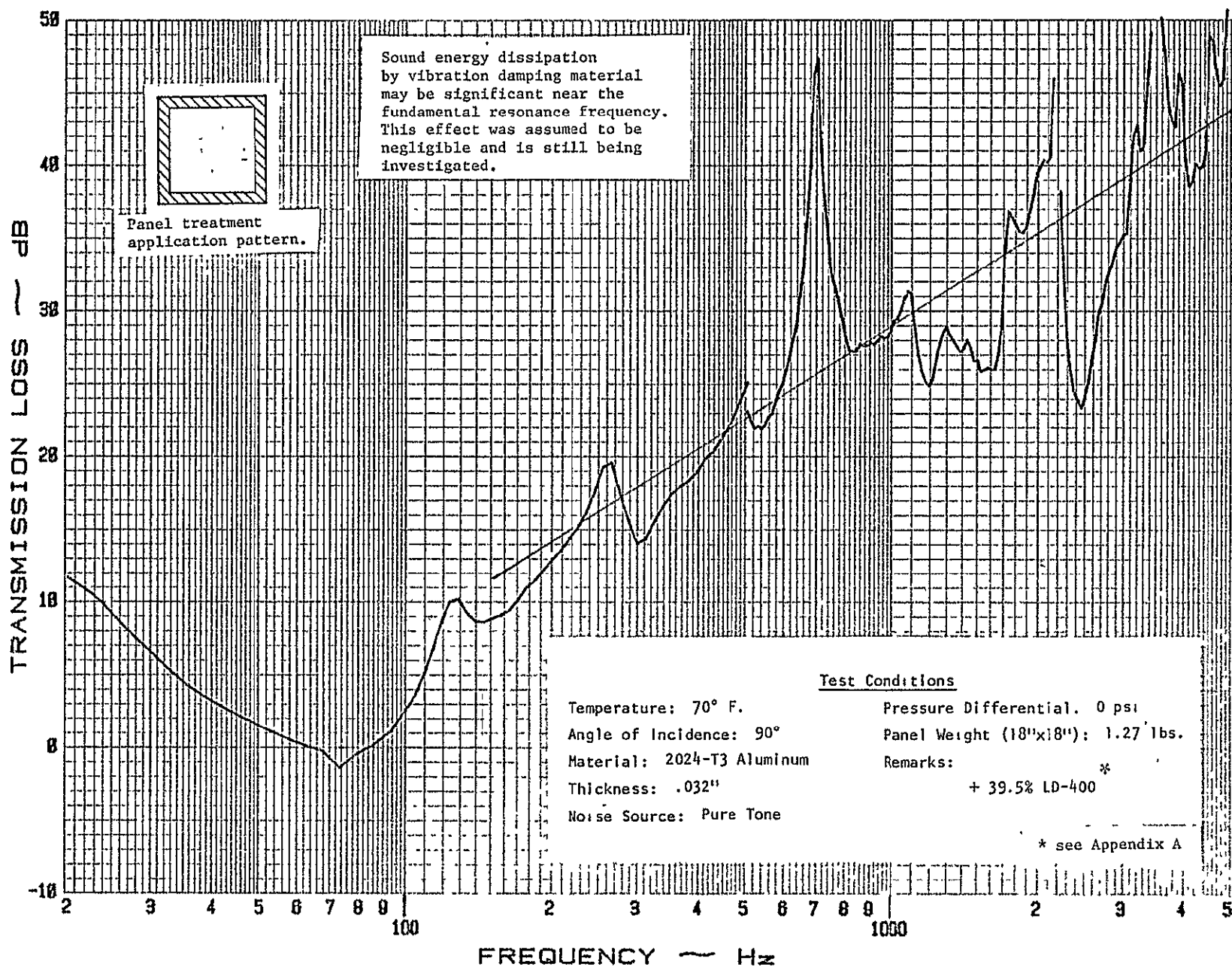
CALC			REVISED	DATE	Figure 7.12: Experimental sound transmission loss of an aluminum panel.	42
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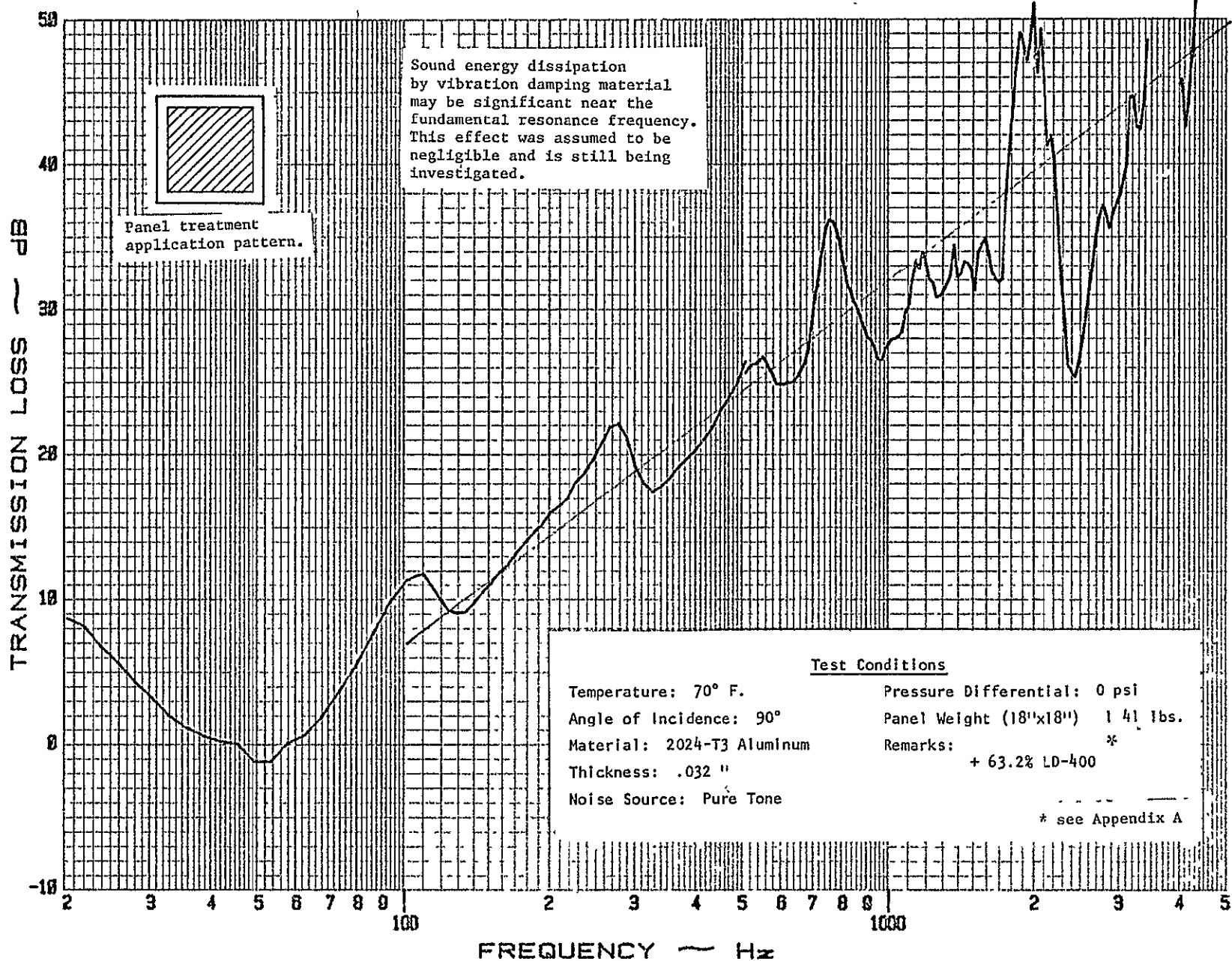


CALC				REVISED	DATE	Figure 7.14: Experimental sound transmission loss of an aluminum panel.	45
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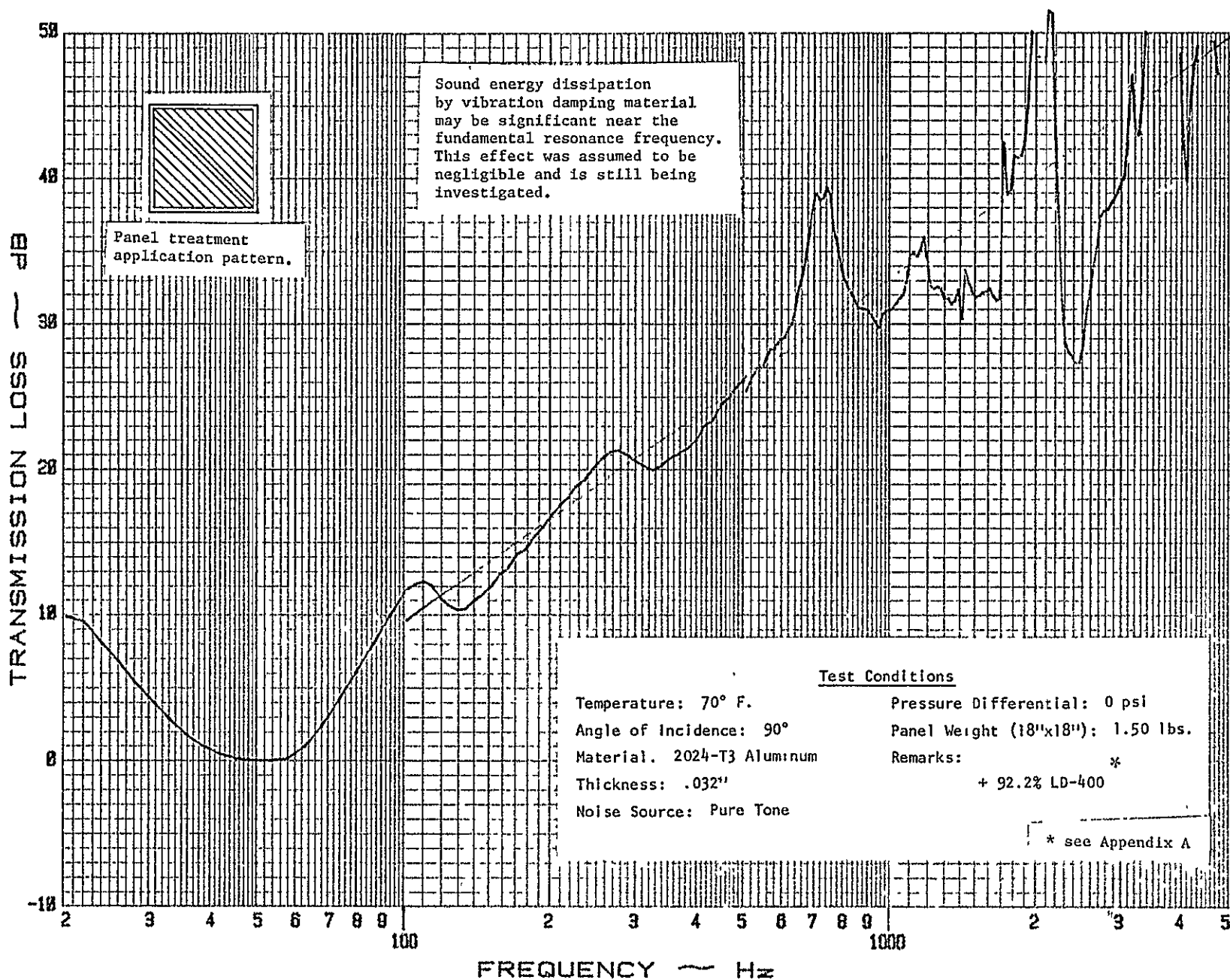


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Figure 7.15: Experimental sound transmission loss of an aluminum panel.	46
PAGE 44	

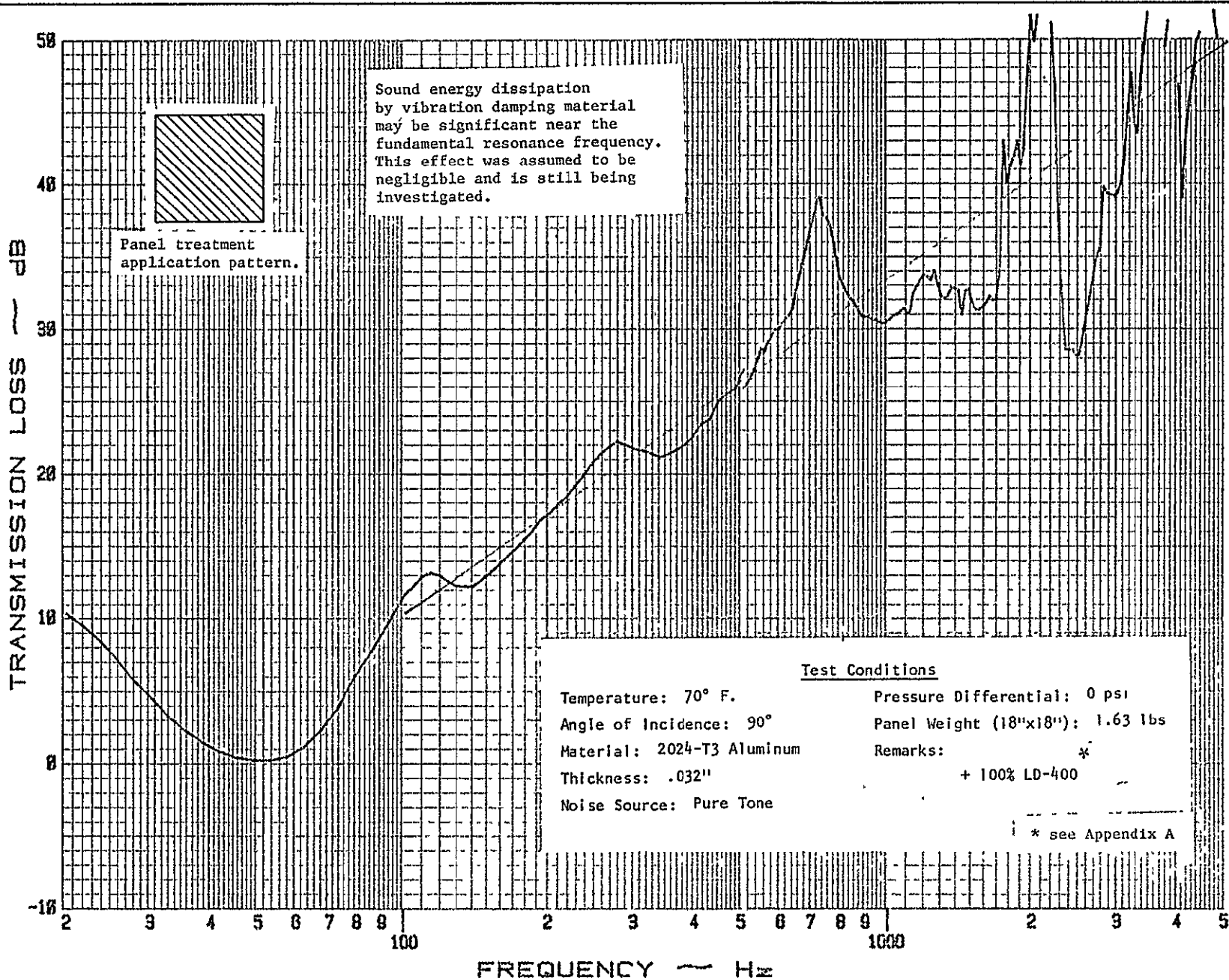


CALC		REVISED	DATE	Figure 7.16: Experimental sound transmission loss of an aluminum panel. UNIVERSITY OF KANSAS
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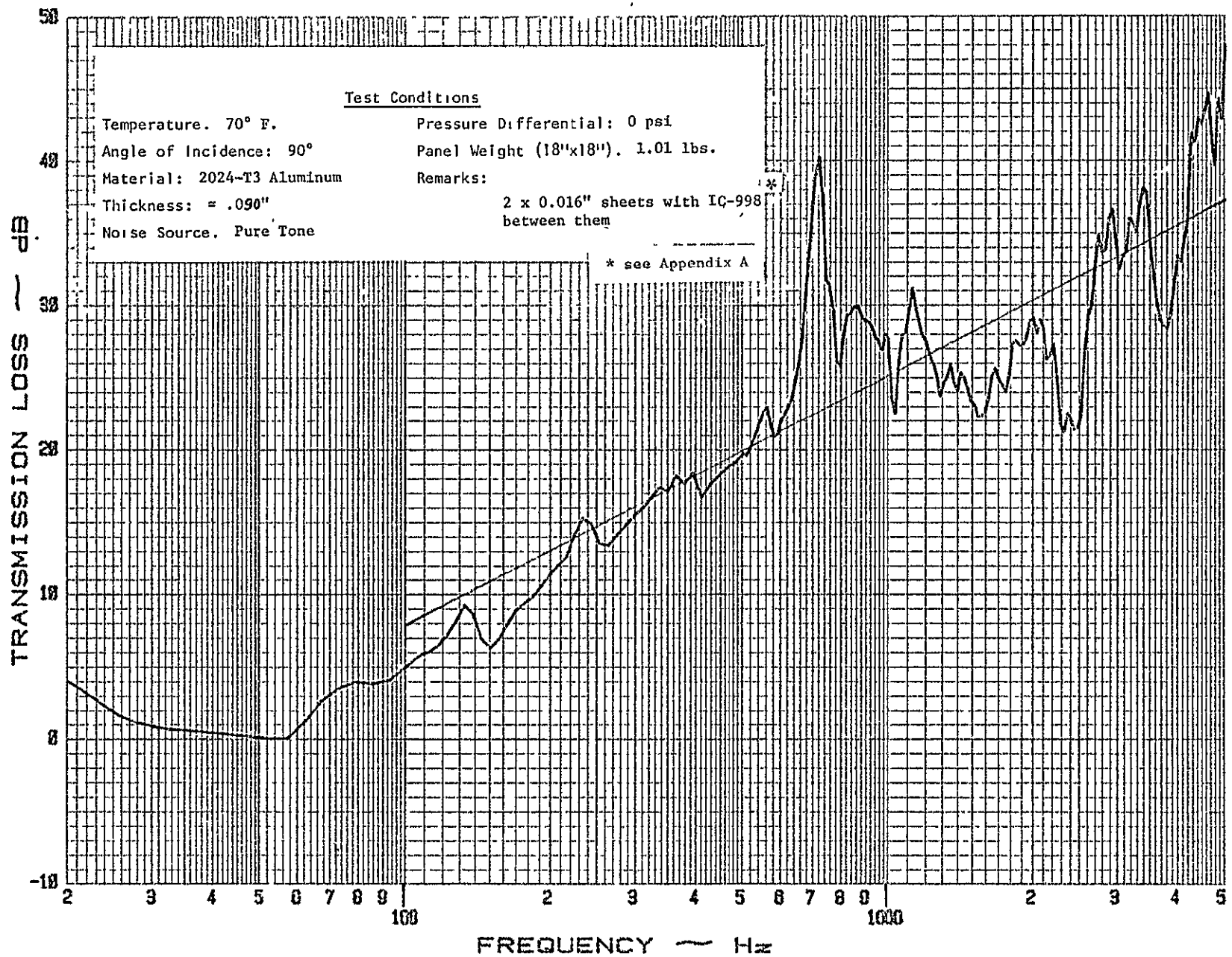


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Figure 7.17: Experimental sound transmission loss of an aluminum panel.

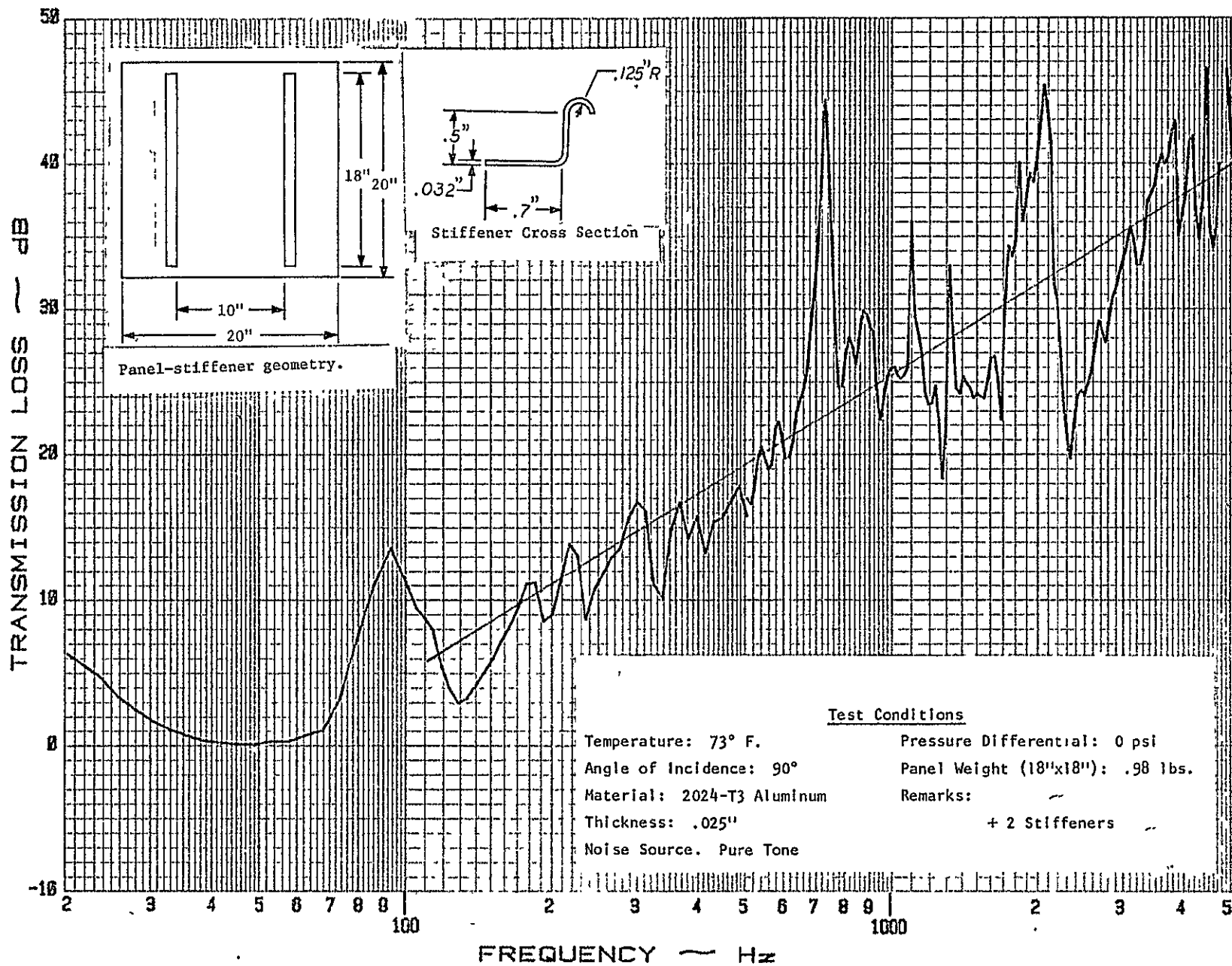


CALC		REVISED	DATE	Figure 7.18: Experimental sound transmission loss of an aluminum panel.	UNIVERSITY OF KANSAS	PAGE 47
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APPD						
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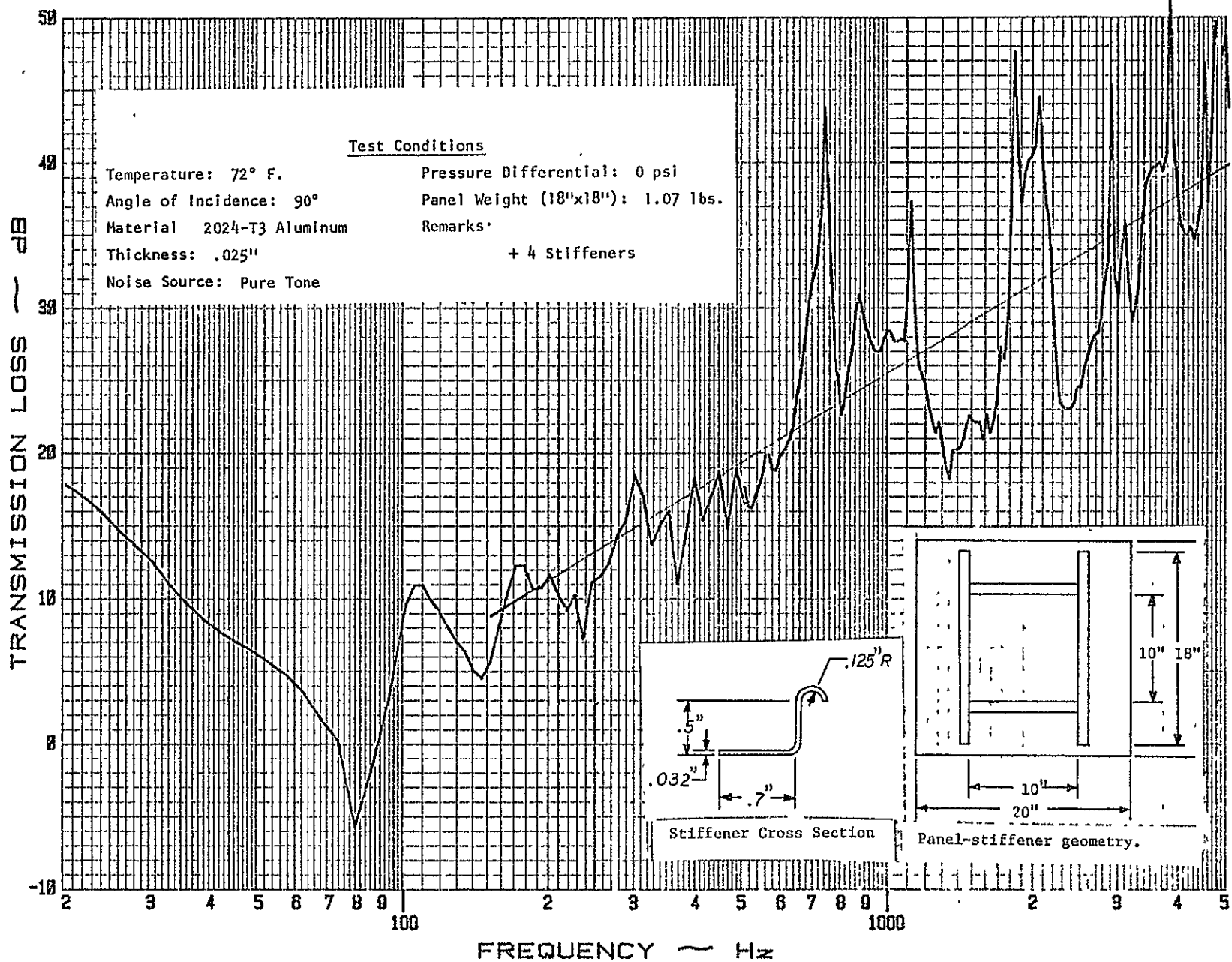


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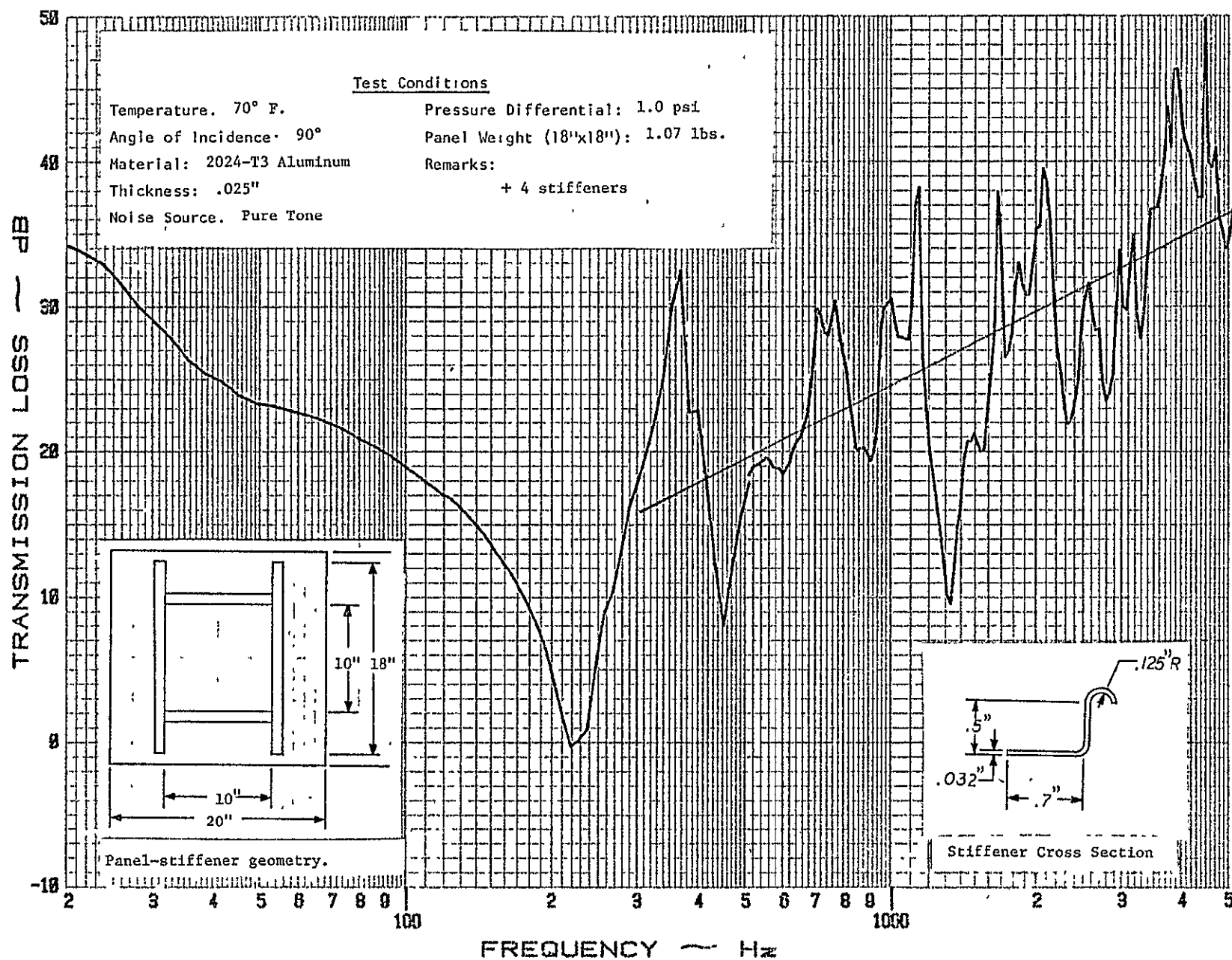
Figure 7.19: Experimental sound transmission loss of an aluminum panel.



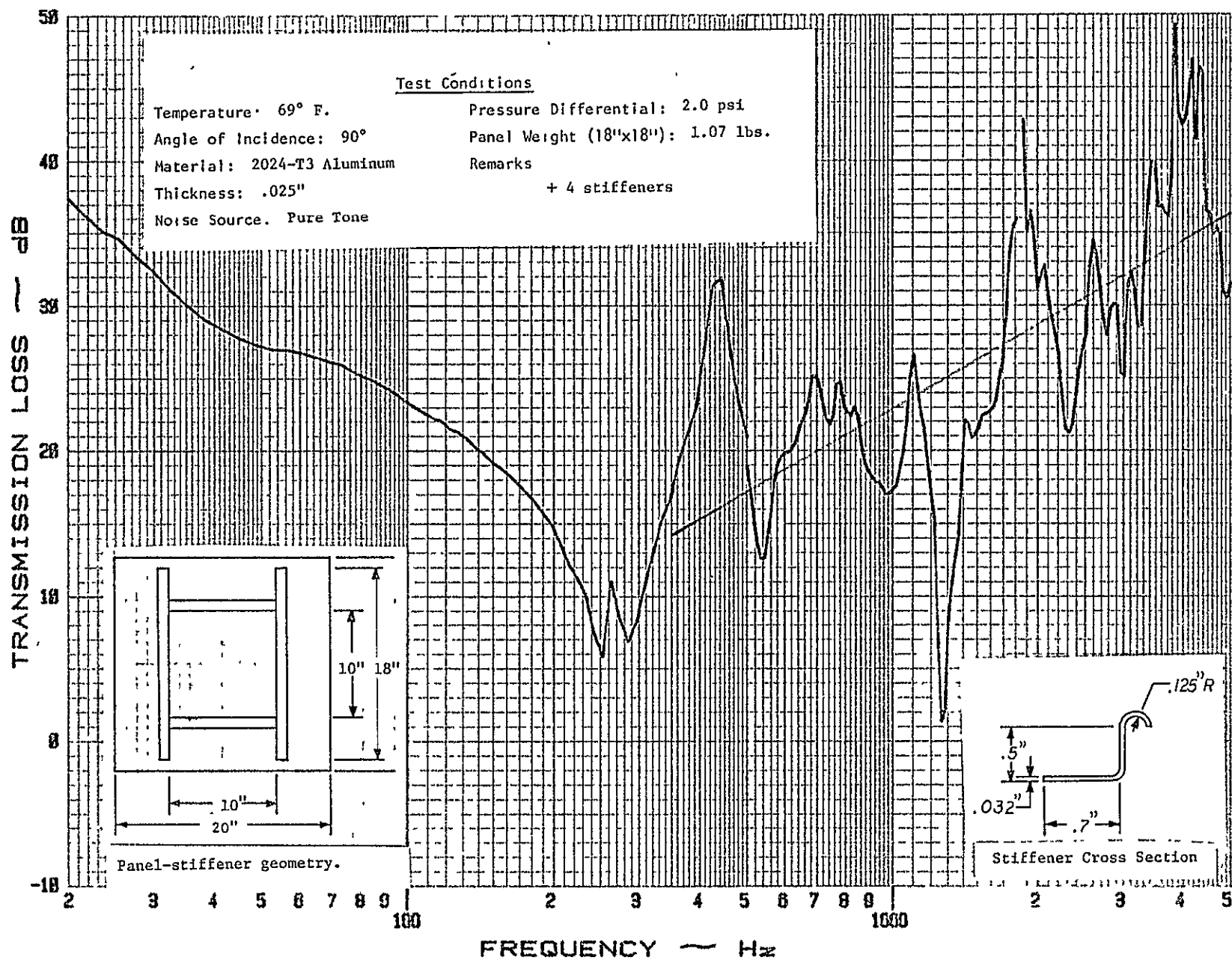
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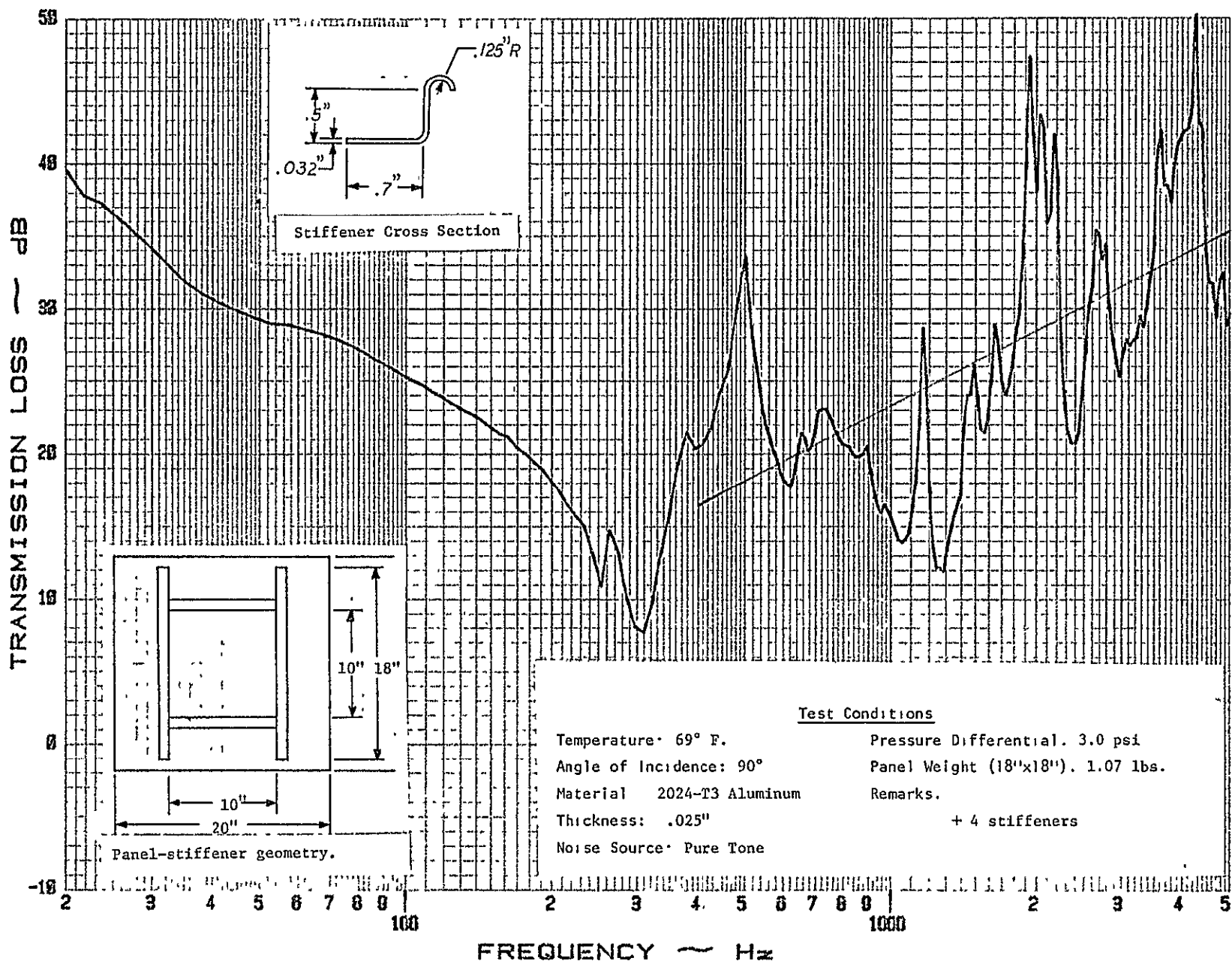
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APPD				UNIVERSITY OF KANSAS	PAGE 50
APPD					



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CHECK						
APPD						
APPD						
UNIVERSITY OF KANSAS					PAGE	51



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APPD								
APPD								
								65

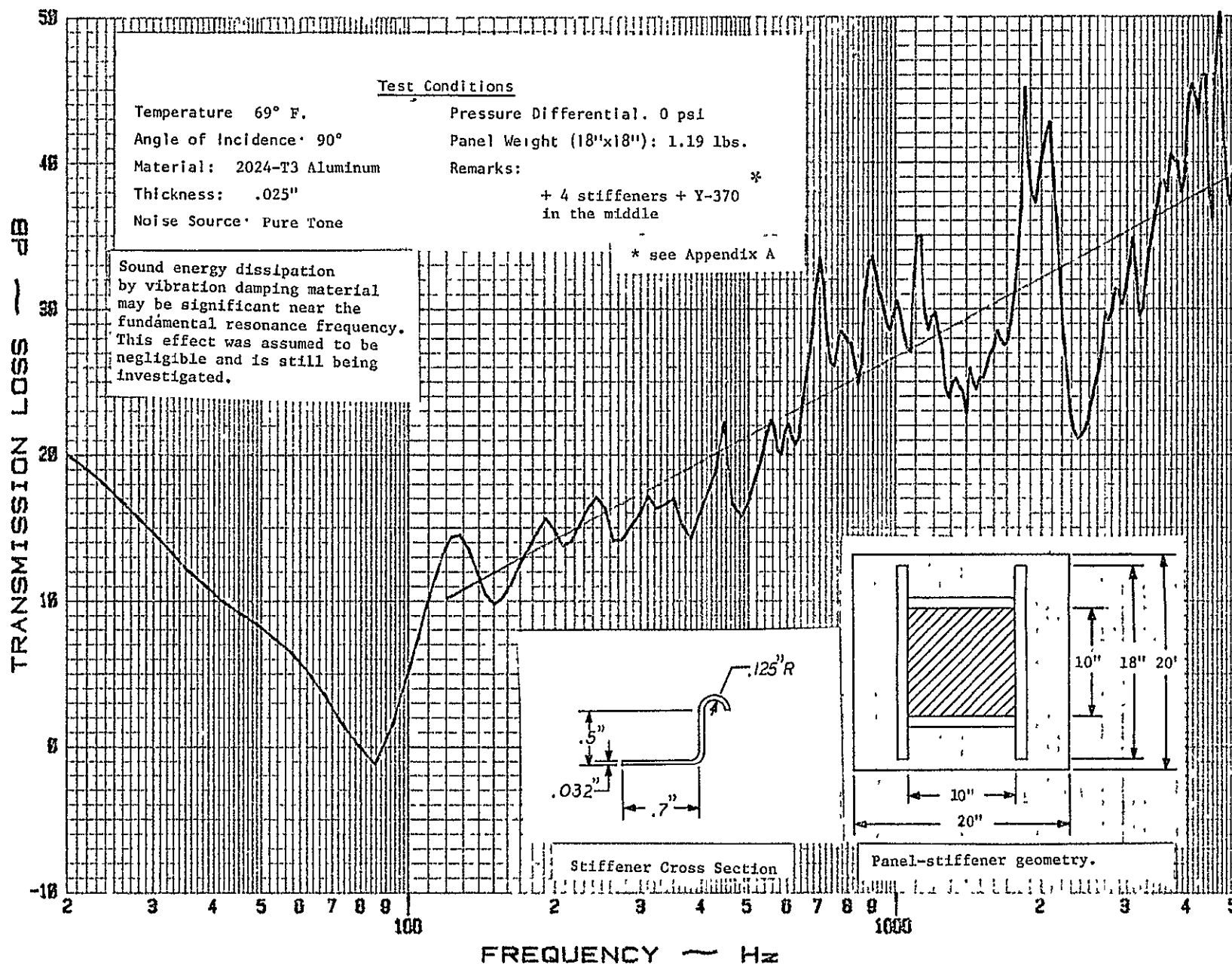


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Figure 7.24: Experimental sound transmission loss of an aluminum panel.

UNIVERSITY OF KANSAS

PAGE 53



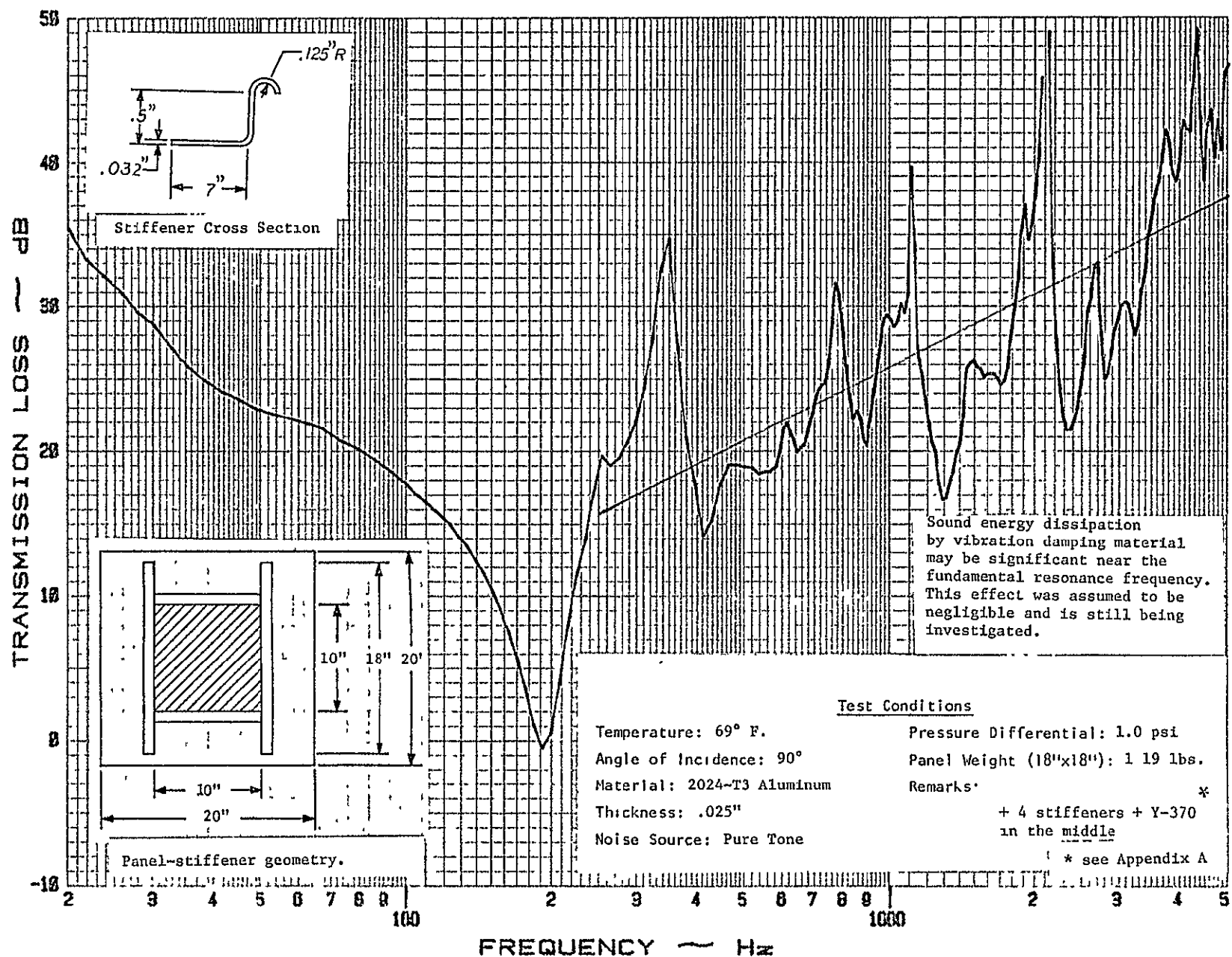


Figure 7.25: Experimental sound transmission loss of an aluminum panel.

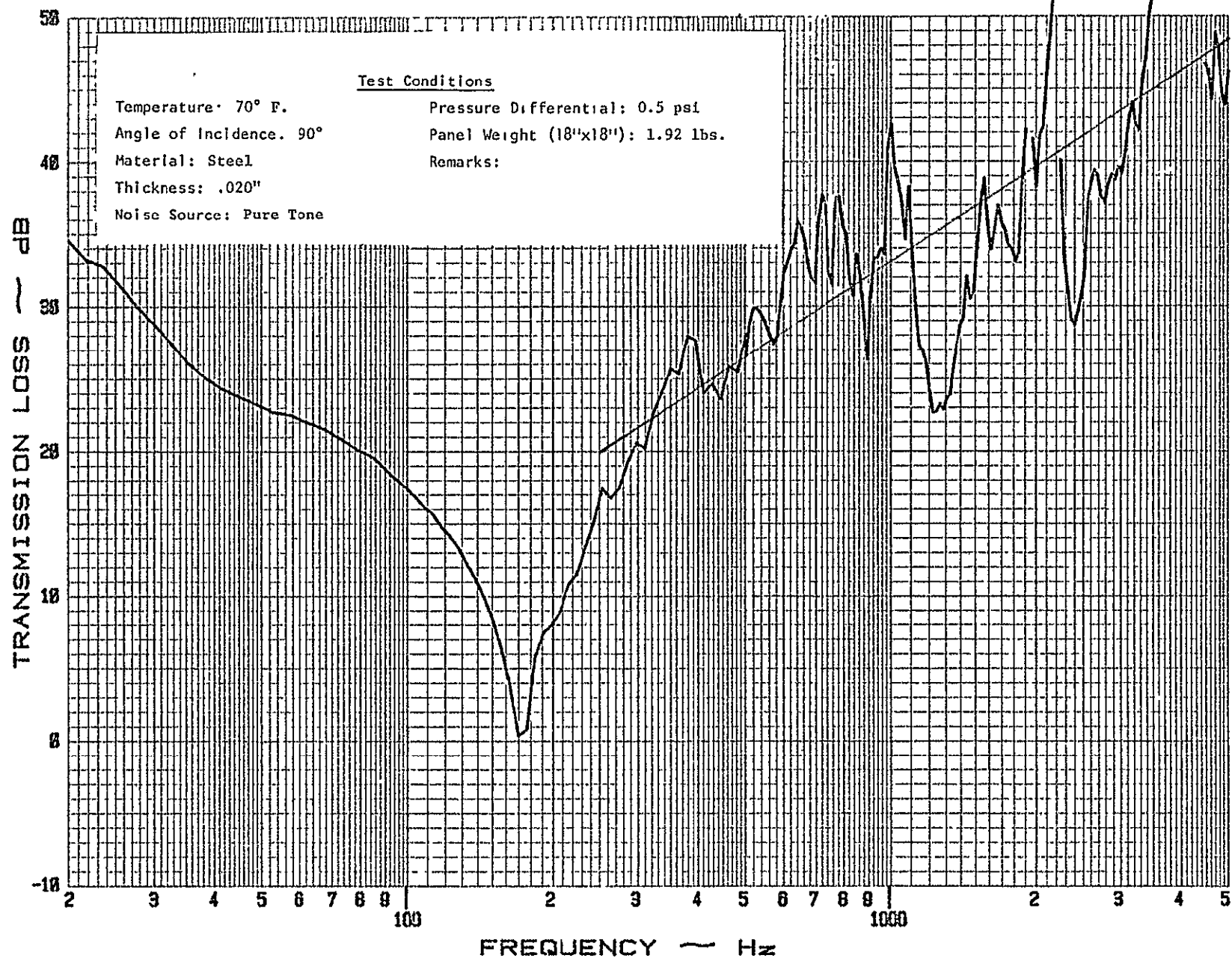


Figure 7.26: Experimental sound transmission loss of a steel panel.

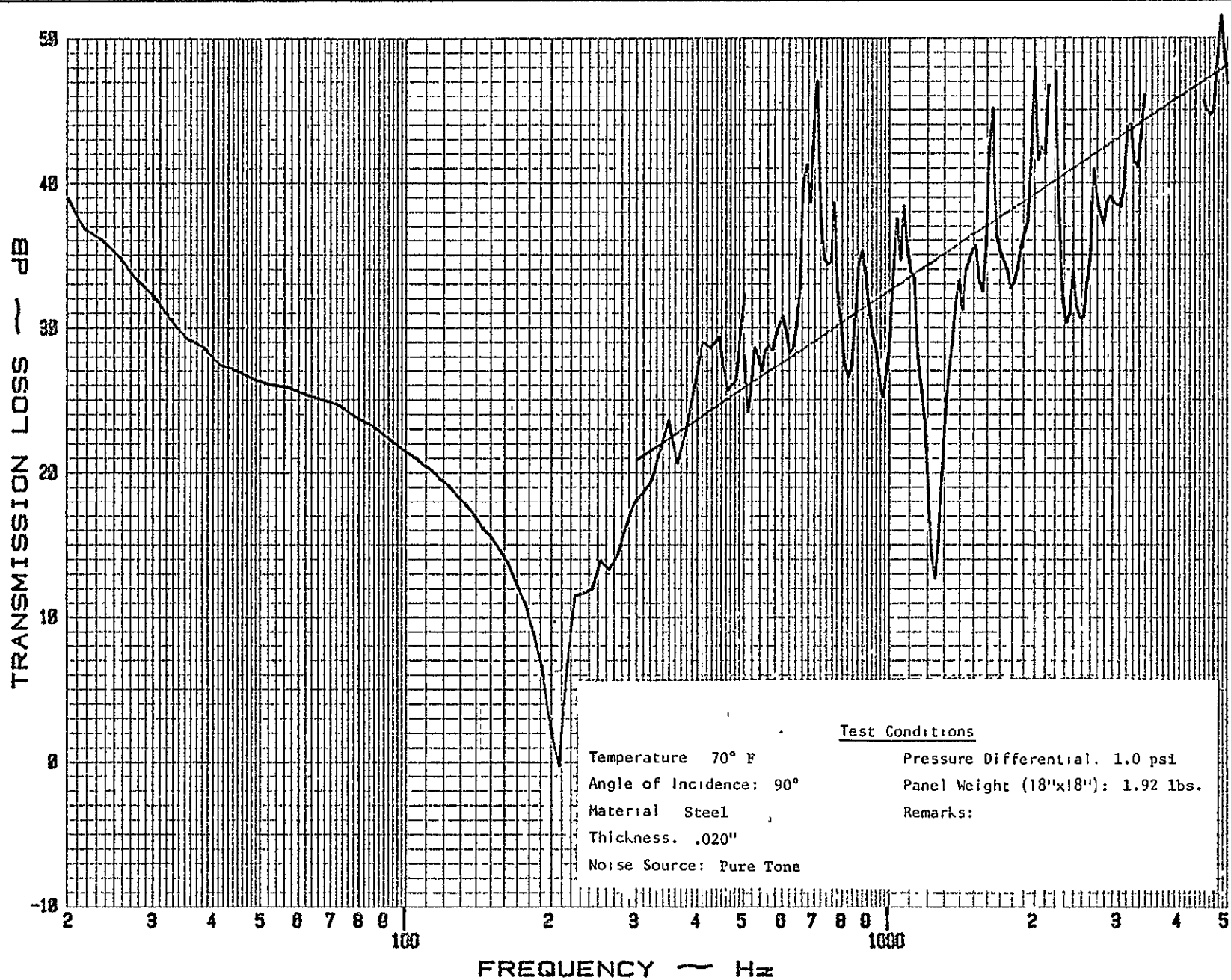


Figure 7.27: Experimental sound transmission loss of a steel panel.

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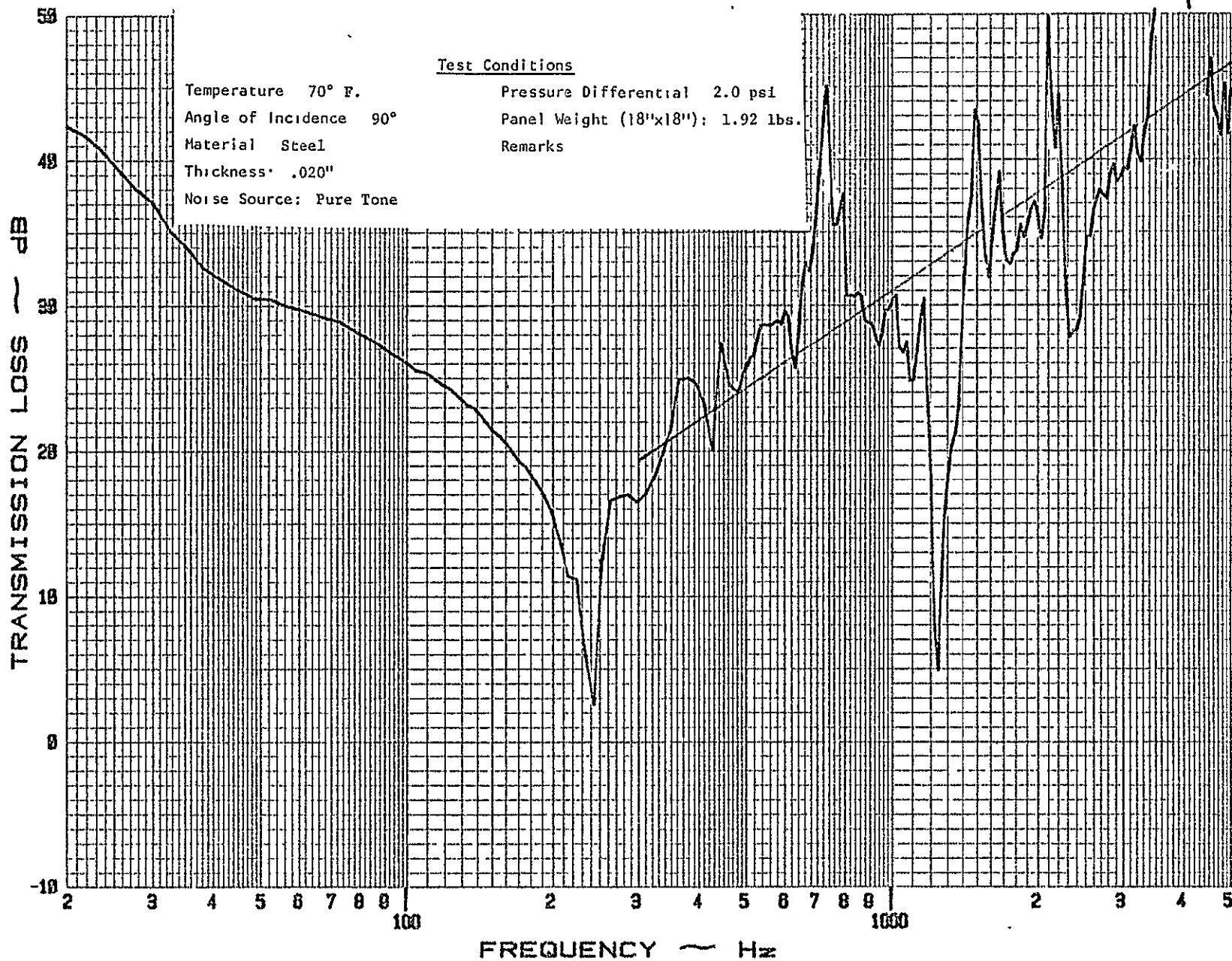
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UNIVERSITY OF KANSAS					PAGE	57



Test Conditions

Temperature: 71° F.

Pressure Differential: 0.5 psi

Angle of Incidence 90°

Panel Weight (18"x18"): .49 lbs.

Material: 2024-T3 Aluminum

Remarks.

Thickness: .016"

Noise Source: Pure Tone

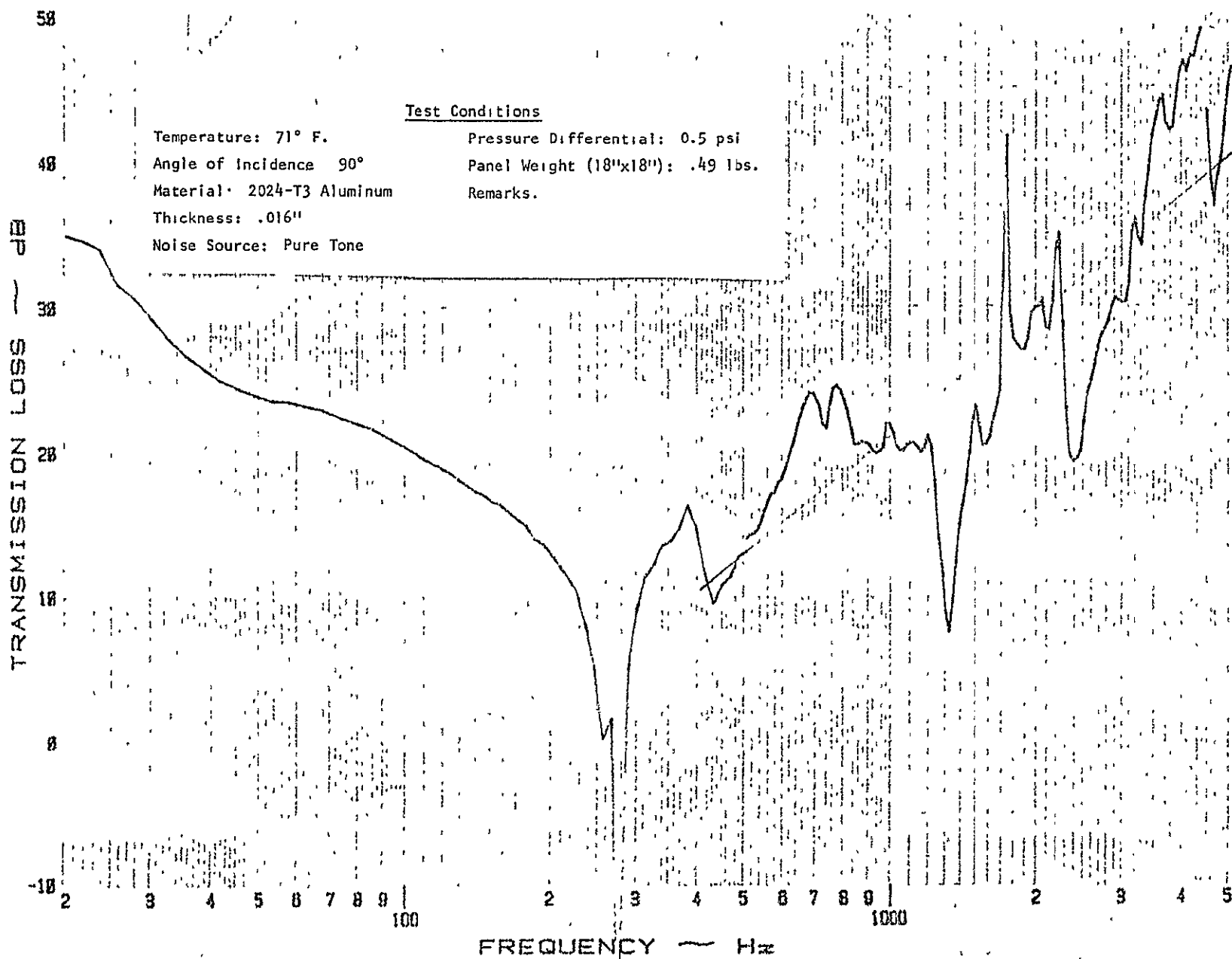


Figure 7.29: Experimental sound transmission loss of an aluminum panel.

UNIVERSITY OF KANSAS

PAGE 58

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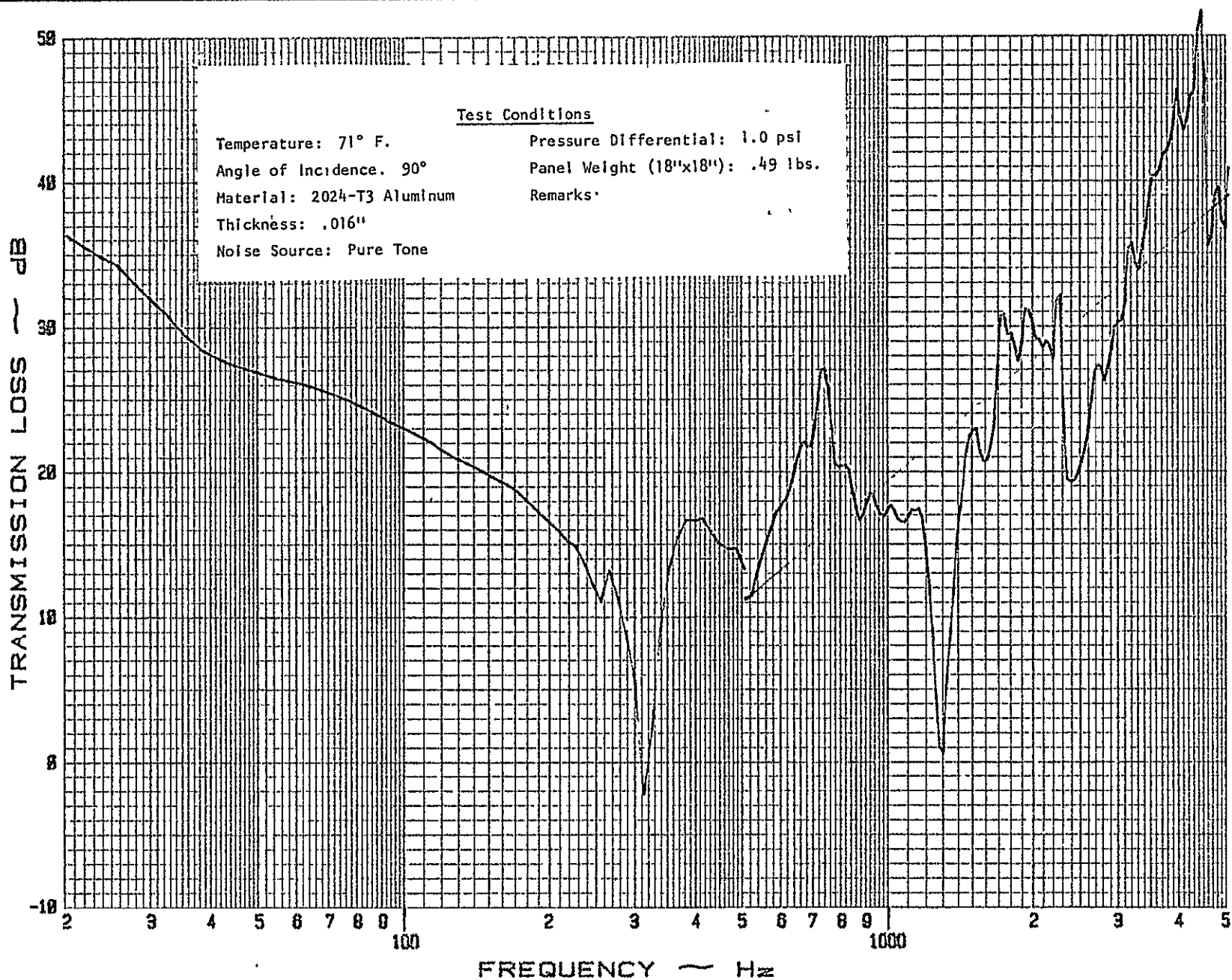


Figure 7.30: Experimental sound transmission loss of an aluminum panel.

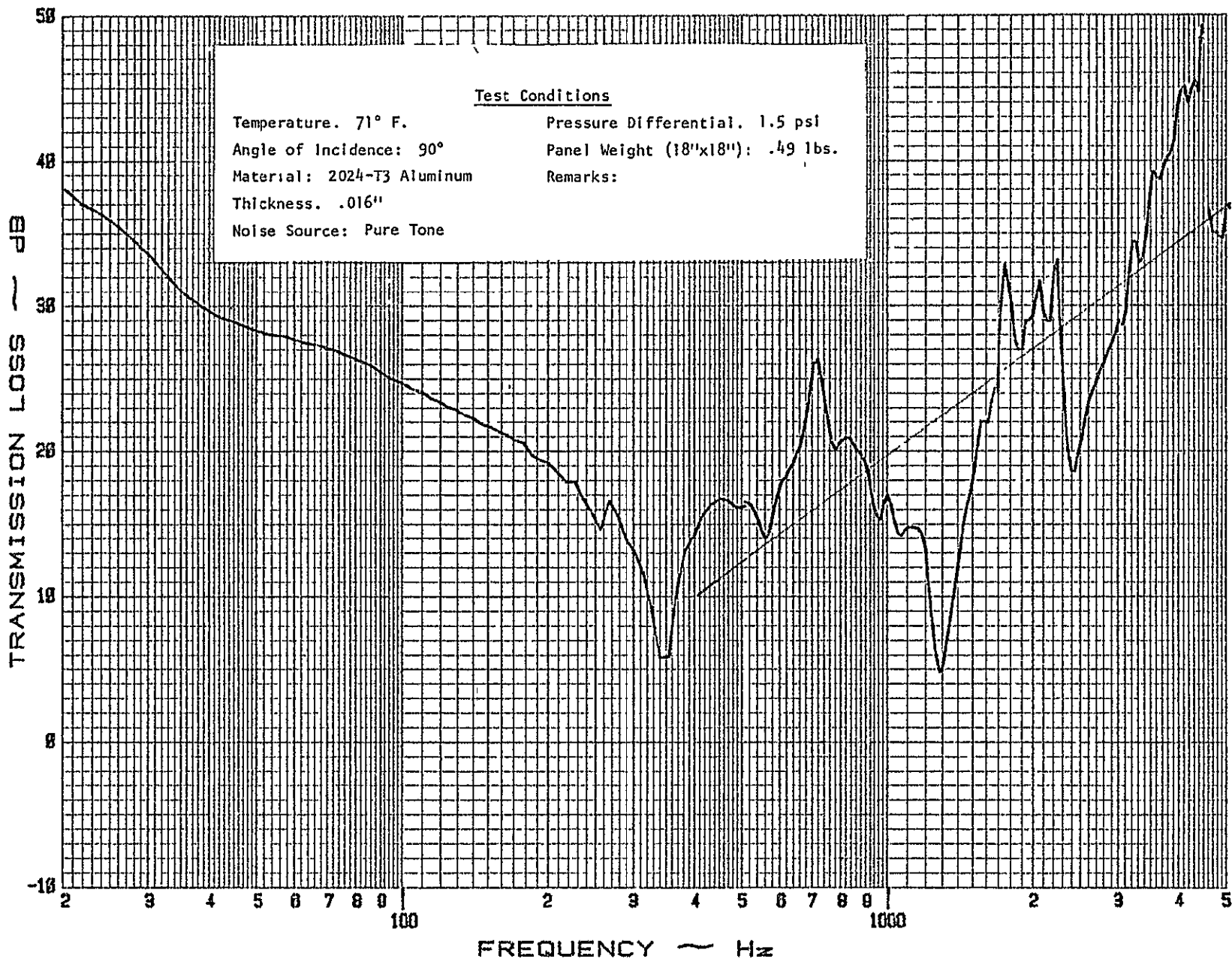


Figure 7.31: Experimental sound transmission loss of an aluminum panel.

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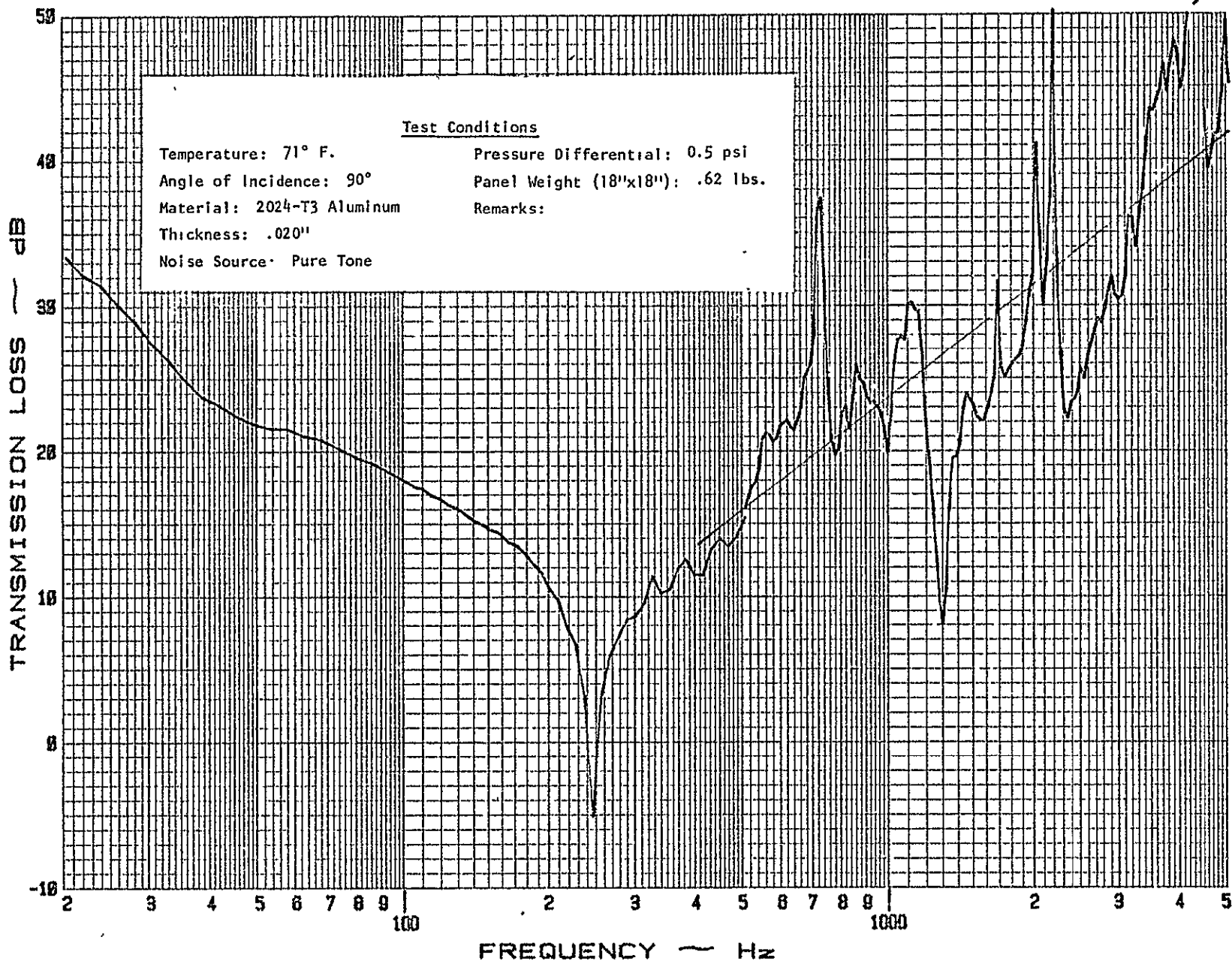
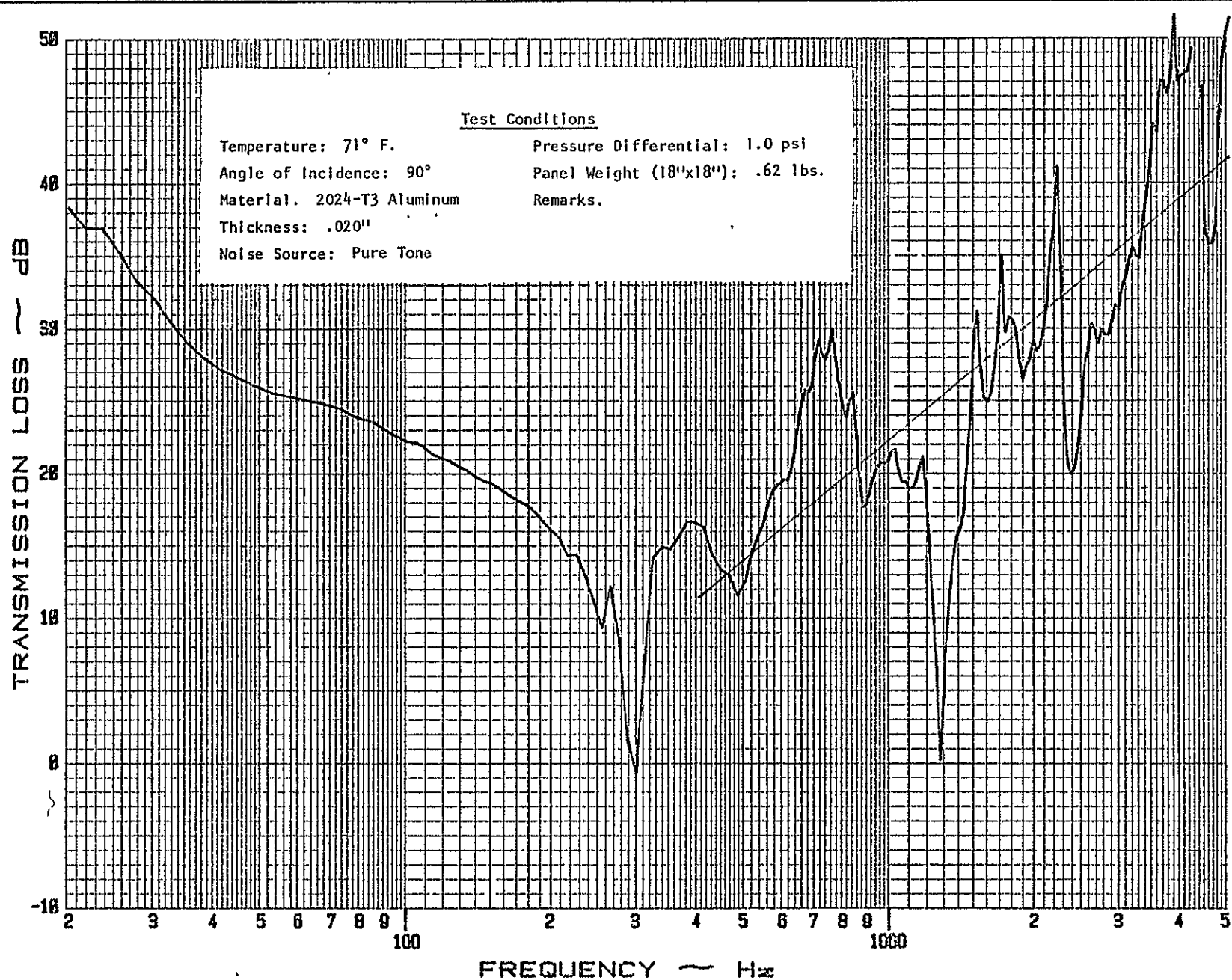


Figure 7.32: Experimental sound transmission loss of an aluminum panel.

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UNIVERSITY OF KANSAS						PAGE 62



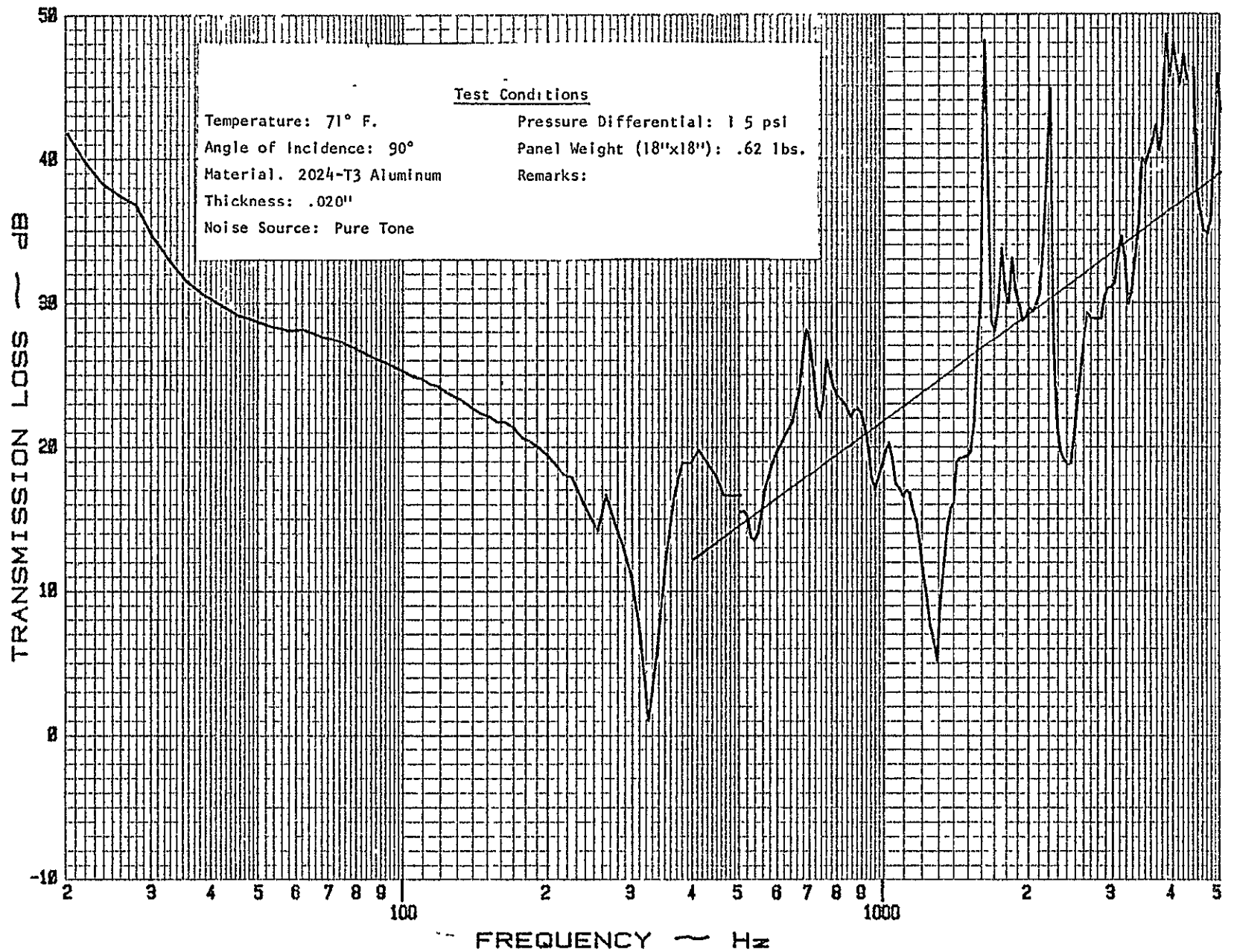


Figure 7.34: Experimental sound transmission loss of an aluminum panel.

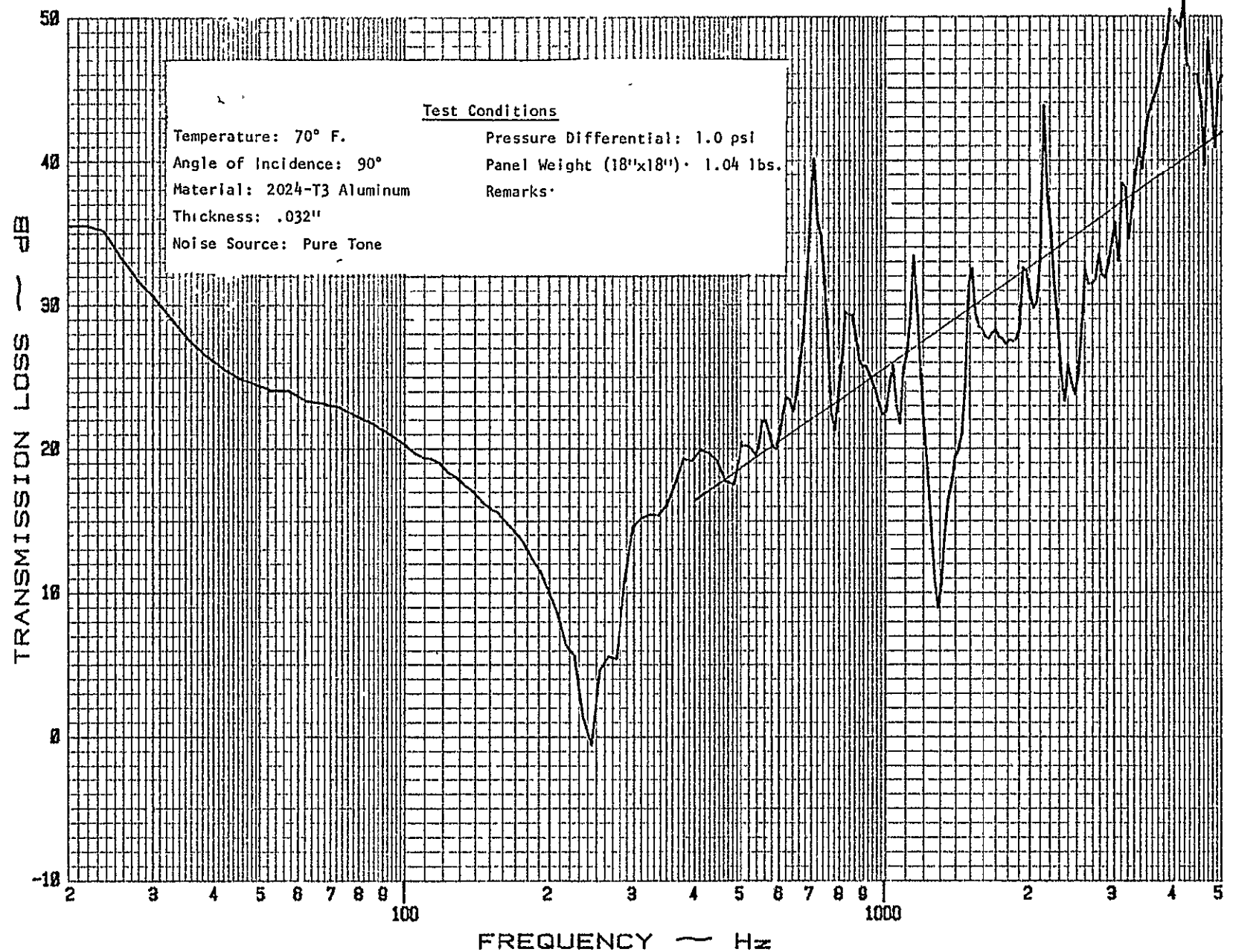


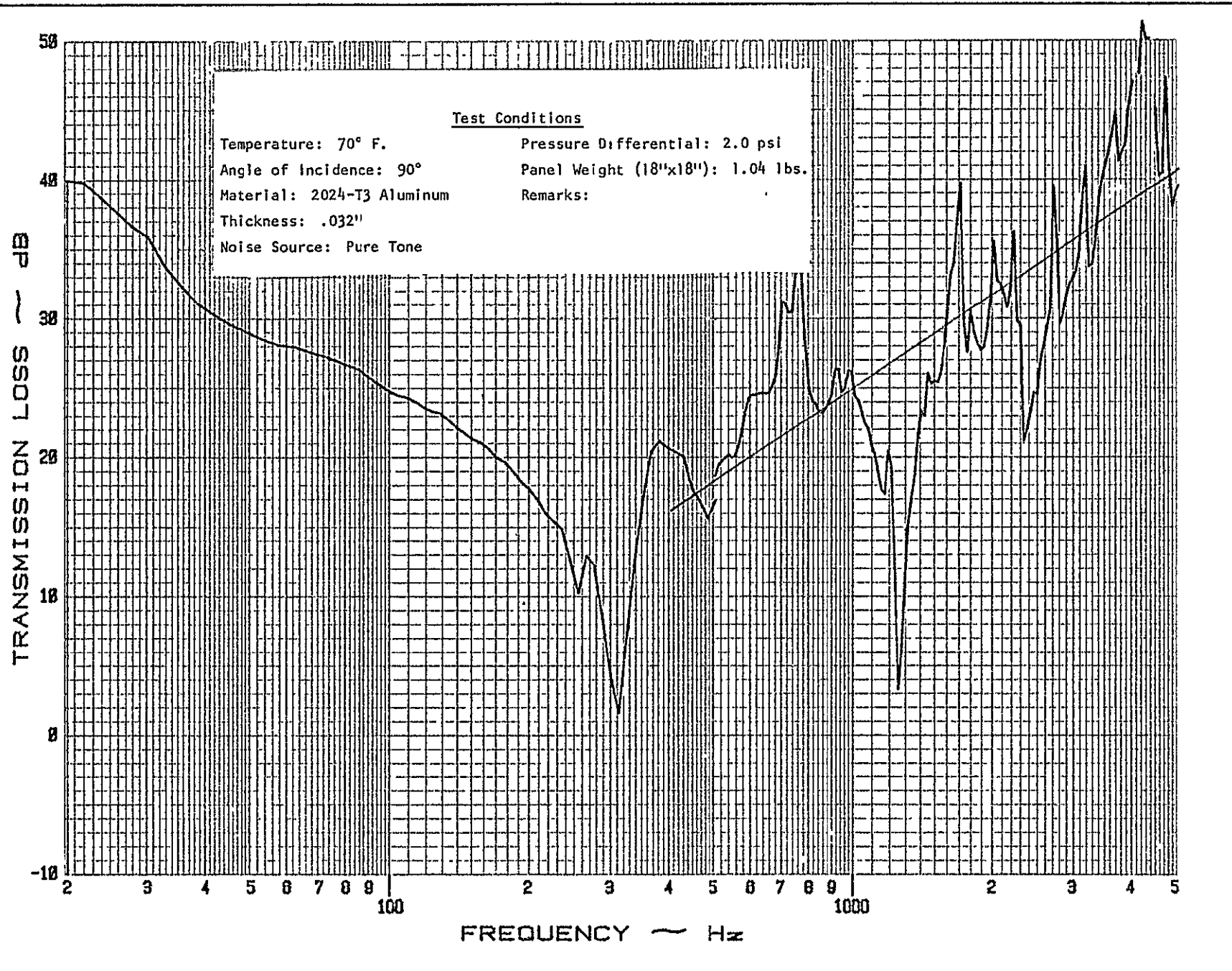
Figure 7.35: Experimental sound transmission loss of an aluminum panel.

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Figure 7.36: Experimental sound transmission loss of an aluminum panel.

UNIVERSITY OF KANSAS

PAGE 65



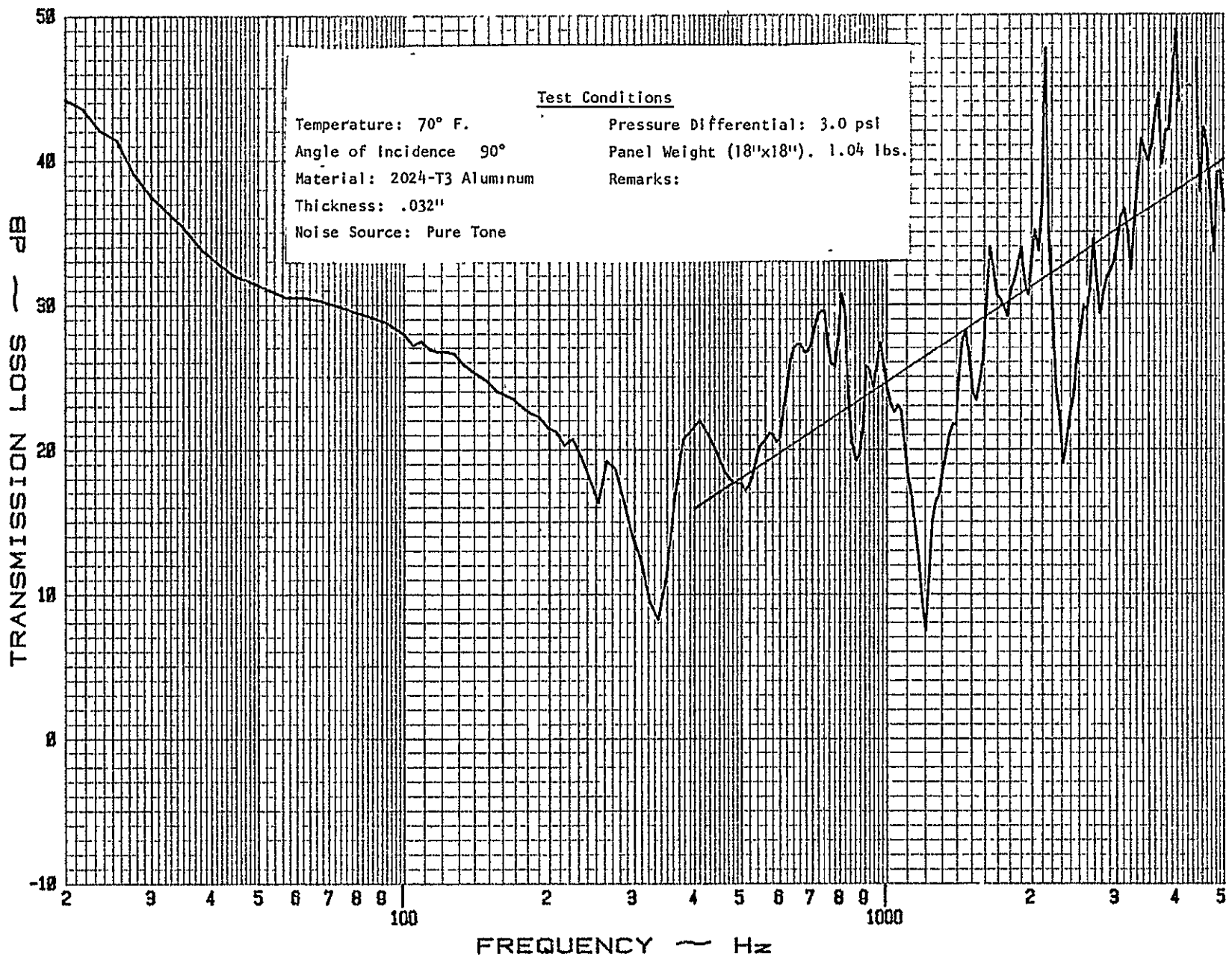
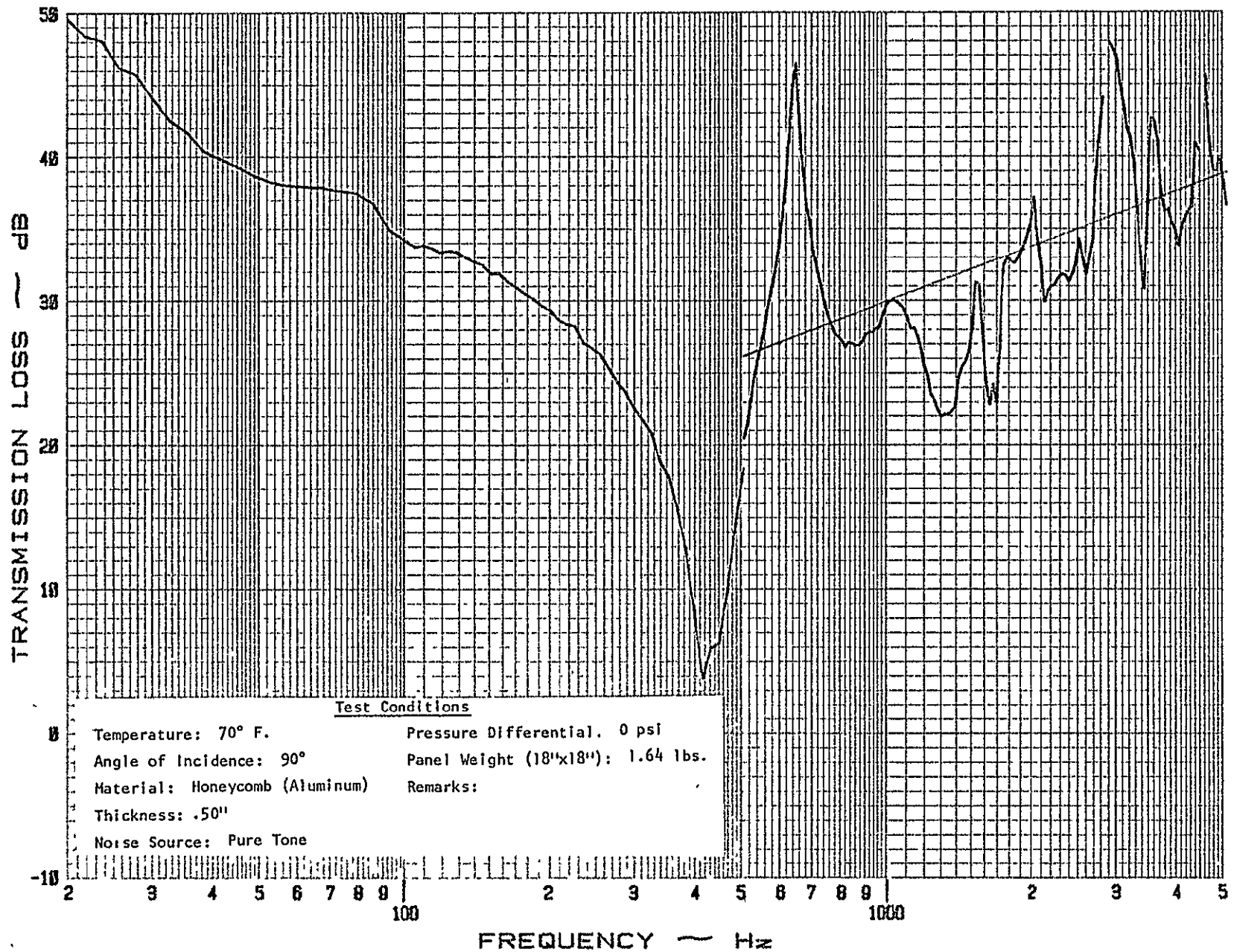


Figure 7.37: Experimental sound transmission loss of aluminum panel.

CALC				REVISD	DATE	UNIVERSITY OF KANSAS	PAGE
CHECK							59
APPD							
APPD							66



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REVISED

DATE

Figure 7.38: Experimental sound trans-

mission loss of a honeycomb aluminum

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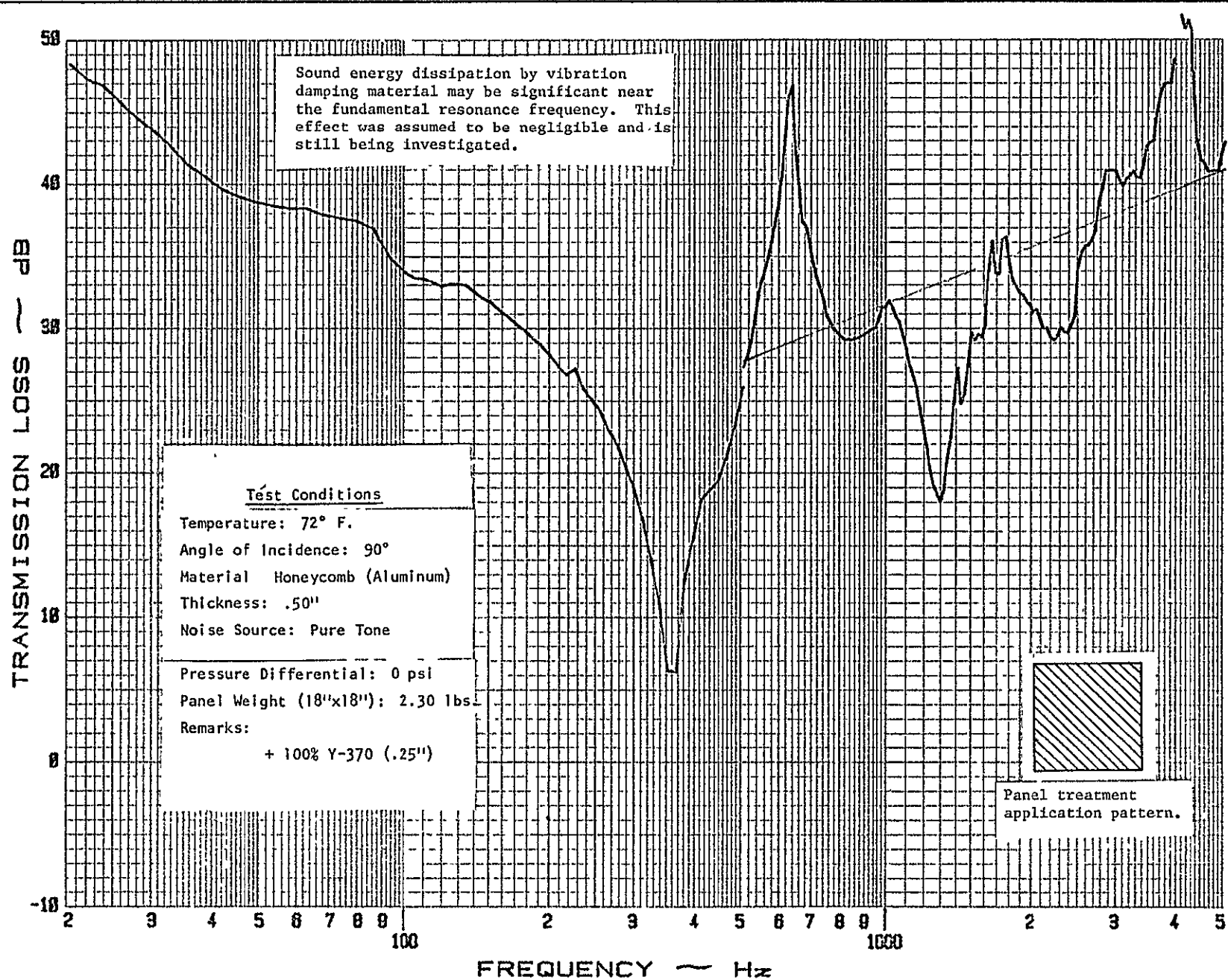
panel.

49

APPD

UNIVERSITY OF KANSAS

PAGE 67



CALC
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APPD
APPD

REVISED

DATE

Figure 7.39: Experimental sound transmission loss of a honeycomb aluminum panel.

UNIVERSITY OF KANSAS

PAGE 68

50

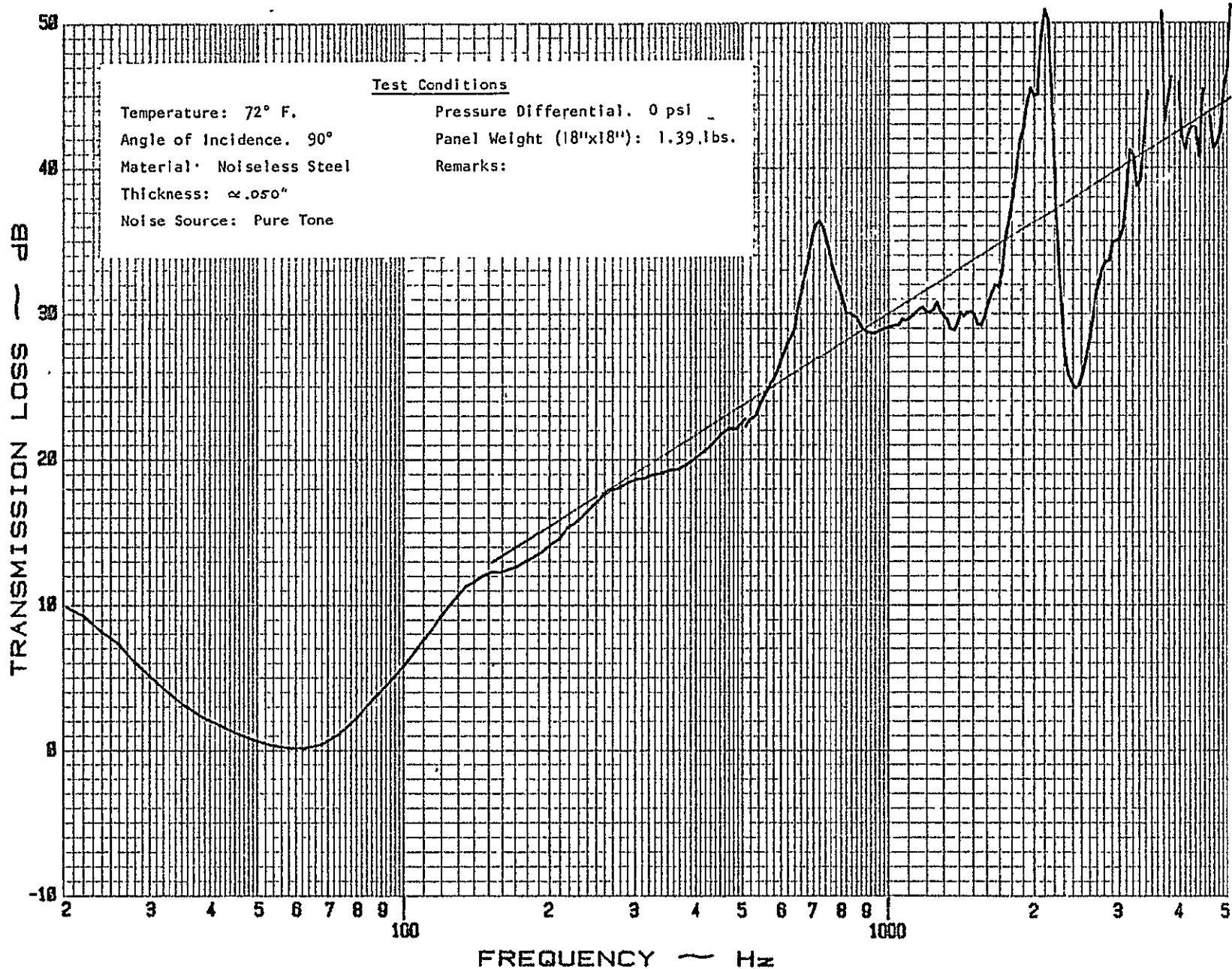


Figure 7.40: Experimental sound transmission loss of a "noiseless steel" panel.

8. Conclusions

The panel test results presented in this report follow predicted trends and generally agree with predicted values of sound transmission loss as well. That is, the well known mass law gives results close to experiment, in the applicable frequency range. An improved stiffness law , which agrees with experiment more closely than do previous stiffness laws, is given in the report.

Slight errors in resonant frequency predictions are thought due to less-than-ideal experimental edge conditions. This is not considered a serious drawback, since actual aircraft panels are likewise "imperfectly" clamped.

Pressurization of panels is found to dramatically increase transmission loss in the low frequency stiffness controlled region. The mechanism responsible is thought to be increased internal membrane stresses; a semi-empirical equation is presented which predicts low frequency transmission loss reasonably well.

References

1. Henderson, T. D., "Design of an Acoustic Panel Test Facility," University of Kansas, M.E. project report, 1977.
2. Franken, P. A., et al., "Methods of Flight Vehicle Noise Prediction," WADC Technical Report 58-343, 1958.
3. Peschier, T. D., "General Aviation Interior Noise Study," University of Kansas, D.E. project report, 1977.
4. Beranek, L. L., Noise and Vibration Control, McGraw-Hill Book Co., New York, 1971.

Appendix A. Vibration Damping Materials

In this appendix some data of the materials applied to panels to prevent vibration will be given.

IC-998

This product is made by Insul-Coustic/Birma Corporation. It is a visco-elastic bonding compound and is designed for bonding of secondary sheet metal panels as treatment for resonant vibration. IC-998 does not require full coverage, but 1/8 in. beads of material applied in a specific design pattern onto resonating surfaces.

Color: brown

Viscosity: thick paste

Application: pneumatic or hand extrusion equipment

Surface temperature limits: -40° F. to 250° F.

Safety: wet - non-flammable

dry - fire resistant

Conforms to NFPA 90 A.

LD-400

This vibration damping material is produced by Lord Corporation. It is a visco-elastic damping material, and the thickness is $\approx$.030 in. The material has to be glued to the panel.

Y-370

This constrained layer visco-elastic damping tape is made by 3M. It is available in several thicknesses; the present research is conducted with the generally used 1/4 in. thickness.

Noiseless Steel

Noiseless steel is a steel/visco-elastic damper/steel sandwich panel produced by Specialty Composites. The total thickness of this panel is $\approx$.050 inch.

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